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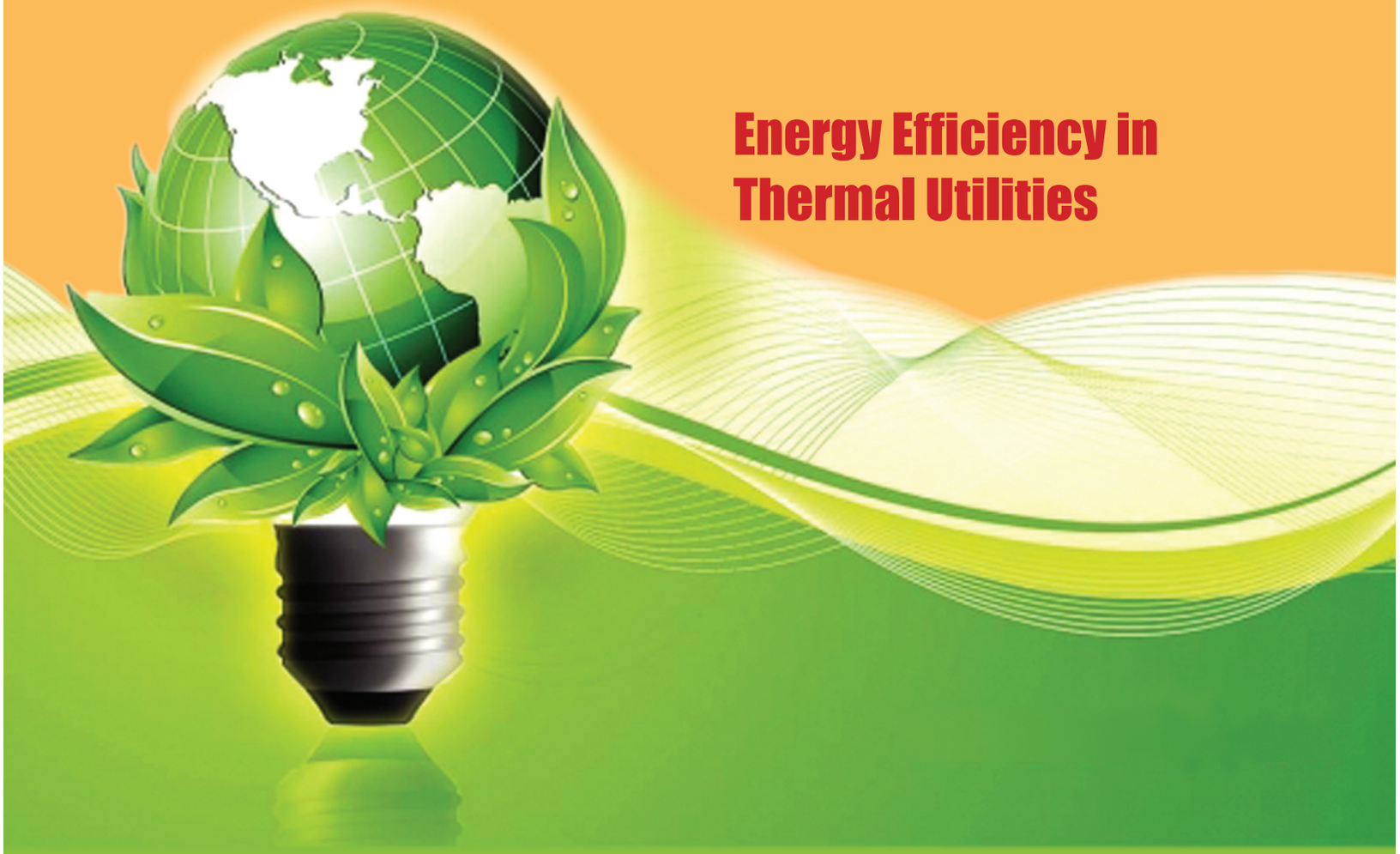

Energy Efficiency in Thermal Utilities

GUIDE BOOK

for

NATIONAL CERTIFICATION EXAMINATION

FOR ENERGY MANAGERS AND ENERGY AUDITORS

Bureau of Energy Efficiency

(A Statutory Body under Ministry of Power, Government of India)

4th Floor, Sewa Bhawan, R. K.Puram, New Delhi - 110 066

www.bee-india.nic.in

FOURTH EDITION, 2015

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PREFACE

National Certification Examination for Energy Managers and Energy Auditors aims to develop qualified professionals in the field of energy conservation in India.

At the initiative of Bureau of Energy Efficiency (BEE) for meeting the requirements of National Certification Examination for Energy Managers and Energy Auditors, Dr. Ambedkar Institute of Productivity, Chennai, the training institute of National Productivity Council of India has prepared four guide books in respect of four examinations as follows:

Guide Book	Examination Paper	Topic
Book-1	Paper-1	General Aspects of Energy Management and Energy Audit
Book-2	**Paper-2**	**Energy Efficiency in Thermal Utilities**
Book-3	Paper-3	Energy Efficiency in Electrical Utilities
Book-4	Paper-4	Energy Performance Assessment for Equipment and Utility Systems

This book-2 on Energy Efficiency in Thermal Utilities focuses on basic principles, guide to efficient operations, performance evaluation techniques of primary fuel using equipment such as boilers, furnaces and cogeneration systems. It also contains guidelines for efficient use of steam and applications of waste heat recovery systems.

Thermal utilities in industry encompass a wide variety of fuels such as oil, coal, gas in addition to different types of agro residues and associated equipment and systems. Thermal energy, constituting a significant portion of the energy bill in many industries, offers huge potential for energy savings.

These books have been prepared with material support from the various books referenced at the end of each chapter. We trust that these books, besides its intended purpose, will also help in enhancing the knowledge base of practicing engineers and managers in understanding and applying energy efficiency techniques.

We hope that this revised edition, incorporating recent developments and practices, will be of use to the prospective energy auditors, managers and other relevant energy efficiency professionals. We invite your valued suggestions and comments, which may be mailed to **guidebooks@em-ea.org**

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March 17, 2015

MESSAGE

India's economic growth depends on the long-term availability of energy from sources that are affordable, accessible and environmentally friendly. Increasing demands for fossil fuels have led to over exploitation of natural resources with adverse impacts on environment and the resultant health issues.

Energy efficiency and its conservation is one of the most cost effective measures to bridge the gap between demand and supply, and to mitigate the environmental impacts of rapid development.

It is heartening to note that the guide books prepared for the National Certification Examination for energy managers and auditors has caught the attention of a wide cross section of the engineering fraternity, not only in India but also in many countries around the world. Many of the techniques and technologies enumerated in these guide books towards enhancing energy efficiency have been put to practice and rich dividends have been realized. Thus these books besides meeting the requirements of the examination are also serving as a manual for practicing engineers.

I commend the efforts put in to prepare such a practical and comprehensive compilation. My best wishes to the users of these books.

(Pradeep K.Sinha)

ENERGY IS LIFE
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(भारत सरकार, विद्युत मंत्रालय)

BUREAU OF ENERGY EFFICIENCY
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MESSAGE

The guide books for National Certification Examination for Energy Managers and Energy Auditors are an enthusiastic celebration of a wide variety of topics related to energy efficiency, especially those which are of practical nature, intricate structure and timely relevance. A glance through the pages of this book will show that it is an unusual type of publication. Though primarily intended for the purpose of being a study material for the examination these guide books have been found to be an out and out a practical guide to implementing energy savings measures in industry and establishments. That they have also been adopted by the energy efficiency promotion organisations of other countries is a testimony to the efficacy of these guide books. Also many of the Indian universities have adopted these books for Post Graduate programmes in energy management.

Written by a group of experienced professionals practicing energy efficiency the books explore the depth and breadth of various possibilities for saving energy in a wide gamut of processes and equipment. Guides such as these are no replacement for the necessary hard work required to realize the potential of energy efficiency but they can certainly help you ensure that your efforts are focused and effective.

I wish all the readers the very best.

(Ajay Mathur)

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ACKNOWLEDGEMENT

Energy efficiency has assumed a critical role in our pursuit of sustainable development. The gains from energy efficiency have been harnessed the world over and India is no exception. Information from the industry and other establishments reveal the rapid strides made in the area of energy efficiency in our country. Examples of industries which have achieved this potential are projected in the energy conservation award programme of Government of India.

As the low hanging fruits appear to have been picked up the path to success in energy management is getting steeper and steeper. More and more information, innovations and knowledge are required to further exploit the energy conservation potential. The continuous efforts to upgrade and refine the guide books for National Certification Examination for Energy Managers and Energy Auditors are aimed at equipping the reader with the resources to identify and implement various energy conservation measures. Through the system of examination we have been able to build sufficient skilled manpower in the area of energy efficiency. We are pleasantly surprised by the enormous interest evinced in the examination from all corners

The unwavering support for the development of these guide books and constructive comments from various quarters about their contents, have been invaluable.

(Bhaskar J. Sarma)

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Chapter 1

Fuels & Combustion

Learning Objectives

In this chapter you will learn about

- Importance of fuel properties
- Basics of combustion
- Characteristics of oil, coal, gas and biomass combustion
- Types of draft

1. FUELS AND COMBUSTION

Introduction to Fuels, Properties of Liquid Fuels, Coal, Gas and Agro Residues, Storage, Handling and Preparation of fuels, Principle of Combustion, Combustion of Oil, Coal, Gas and Biomass, Draft System, Combustion Controls.

1.1 Introduction to Fuels

The various types of fuels like liquid, solid and gaseous fuels are available for firing in boilers, furnaces and other combustion equipments. The selection of right type of fuel depends on various factors such as availability, storage, handling, pollution and landed cost of fuel.

The knowledge of the fuel properties helps in selecting the right fuel for the right purpose and efficient use of the fuel. The following characteristics, determined by laboratory tests, are generally used for assessing the nature and quality of fuels.

1.2 Properties of Liquid Fuels

Liquid fuels like furnace oil and LSHS are predominantly used in industrial application. The various properties of liquid fuels are given below.

Density

This is defined as the ratio of the mass of the fuel to the volume of the fuel at a reference temperature of 15°C. Density is measured by an instrument called hydrometer. The knowledge of density is useful for quantity calculations and assessing ignition quality. The unit of density is kg/m^3.

Specific gravity

This is defined as the ratio of the weight of a given volume of oil to the weight of the same volume of water at a given temperature. The density of fuel, relative to water, is called specific gravity. The specific gravity of water is defined as 1. Since specific gravity is a ratio, it has no units. The measurement of specific gravity is generally made by a hydrometer.

Specific gravity is used in calculations involving weights and volumes. The specific gravity of various fuel oils are given in Table 1.1

Table 1.1 Specific Gravity of Various Fuel Oils

Fuel Oil	**L.D.O** Light Diesel Oil	**Furnace oil**	**L.S.H.S** Low Sulphur Heavy Stock
Specific Gravity	0.85-0.87	0.89-0.95	0.88-0.98

Viscosity

The viscosity of a fluid is a measure of its internal resistance to flow. Viscosity depends on temperature and decreases as the temperature increases. Any numerical value for viscosity has no meaning unless

the temperature is also specified. Viscosity is measured in Stokes / Centistokes. Sometimes viscosity is also quoted in Engler, Saybolt or Redwood. Each type of oil has its own temperature - viscosity relationship. The measurement of viscosity is made with an instrument called Viscometer.

Viscosity is the most important characteristic in the storage and use of fuel oil. It influences the degree of pre-heat required for handling, storage and satisfactory atomization. If the oil is too viscous, it may become difficult to pump, hard to light the burner, and tough to operate. Poor atomization may result in the formation of carbon deposits on the burner tips or on the walls. Therefore pre-heating is necessary for proper atomization.

Flash Point

The flash point of a fuel is the lowest temperature at which the fuel can be heated so that the vapour gives off flashes momentarily when an open flame is passed over it. Flash point for furnace oil is 66^{o}C.

Pour Point

The pour point of a fuel is the lowest temperature at which it will pour or flow when cooled under prescribed conditions. It is a very rough indication of the lowest temperature at which fuel oil is readily pumpable

Specific Heat

Specific heat is the amount of kcal needed to raise the temperature of 1 kg of oil by 1^{o}C. The unit of specific heat is kcal/kg^{o}C. It varies from 0.22 to 0.28 depending on the oil specific gravity. The specific heat determines how much steam or electrical energy it takes to heat oil to a desired temperature. Light oils have a low specific heat, whereas heavier oils have a higher specific heat.

Calorific Value

The calorific value is the measurement of heat or energy produced, and is measured either as gross calorific value or net calorific value. The difference being the latent heat of condensation of the water vapour produced during the combustion process. Gross calorific value (GCV) assumes all vapour produced during the combustion process is fully condensed. Net calorific value (NCV) assumes the water leaves with the combustion products without fully being condensed. Fuels should be compared based on the net calorific value.

The calorific value of coal varies considerably depending on the ash, moisture content and the type of coal while calorific value of fuel oil is much more consistent. The typical Gross Calorific Values of some of the commonly used liquid fuels are given below:

Fuel Oil	Gross Calorific Value (kcal/kg)
Kerosene	- 11,100
Diesel Oil	- 10,800
L.D.O	- 10,700
Furnace Oil	- 10,500
LSHS	- 10,600

The following conversion formula shows the difference between GCV and NCV.

$GCV = NCV + 584 ((9H_2\% + M\%)/100)$

Where,

GCV= Gross calorific value of fuel, kcal/kg
NCV = Net calorific value of fuel, kcal/kg
H_2%= Hydrogen % by weight present in the fuel
M% = Moisture % by weight present in the fuel
584 = Latent heat corresponding to partial pressure of water vapour, kcal/kg

Sulphur

The amount of sulphur in the fuel oil depends mainly on the source of the crude oil and to a lesser extent on the refining process. The normal sulfur content for the residual fuel oil (furnace oil) is in the order of 2-4 %.

Typical figures are:

Fuel oil	Percentage of Sulphur
Kerosene	0.05—0.2
Diesel Oil	0.05 – 0.25
L.D.O	0.5 – 1.8
Furnace Oil	2.0 – 4.0
LSHS	< 0.5

The main disadvantage of sulphur is the risk of corrosion by sulphuric acid formed during and after combustion, and condensing in cool parts of the chimney or stack, air pre heater and economiser.

Ash Content

The ash value is related to the inorganic material in the fuel oil. The ash levels of distillate fuels are negligible. Residual fuels have more of the ash-forming constituents. These salts may be compounds of sodium, vanadium, calcium, magnesium, silicon, iron, aluminum, nickel, etc.

Typically, the ash value is in the range 0.03-0.07 %. Excessive ash in liquid fuels can cause fouling deposits in the combustion equipment. Ash has erosive effect on the burner tips, causes damage to the refractories at high temperatures and gives rise to high temperature corrosion and fouling of equipments.

Carbon Residue

Carbon residue indicates the tendency of oil to deposit a carbonaceous solid residue on a hot surface, such as a burner or injection nozzle, when its vaporisable constituents evaporate. Residual oil contains carbon residue ranging from 1 percent or more.

Water Content

Water content of furnace oil when supplied is normally very low as the product at refinery site is handled hot and maximum limit of 1% is specified in the standard.

Water may be present in free or emulsified form and can cause damage to the inside furnace surfaces during combustion especially if it contains dissolved salts. It can also cause spluttering of the flame at the burner tip, possibly extinguishing the flame and reducing the flame temperature or lengthening the flame.

Typical specification of fuel oil is summarised in the Table 1.2.

Table 1.2 Typical Specification of Fuel Oils

Properties	Fuel Oils		
	Furnace Oil	**LS.H.S.**	**L.D.O.**
Density (Approx. g/cc at 15 ^{0}C)	0.89-0.95	0.88-0.98	0.85-0.87
Flash Point (^{0}C)	66	93	66
Pour Point (^{0}C)	20	72	18
G.C.V. (kcal/kg)	10,500	10,600	10,700
Sediment, % Wt. Max.	0.25	0.25	0.1
Sulphur Total, % Wt. Max.	Up to 4.0	Up to 0.5	Up to 1.8
Water Content, % Vol. Max.	1.0	1.0	0.25
Ash % Wt. Max.	0.1	0.1	0.02

Storage of Fuel oil

It can be potentially hazardous to store furnace oil in barrels. A better practice is to store it in cylindrical tanks, either above or below the ground. Furnace oil, that is delivered, may contain dust, water and other contaminants.

The sizing of storage tank facility is very important. A recommended storage estimate is to provide for at least 10 days of normal consumption. Industrial heating fuel storage tanks are generally vertical mild steel tanks mounted above ground. It is prudent for safety and environmental reasons to build bund walls around tanks to contain accidental spillages.

As a certain amount of settlement of solids and sludge will occur in tanks over time, cleaning should be carried out at regular intervals-annually for heavy fuels and every two years for light fuels. A little care should be taken when oil is decanted from the tanker to storage tank. All leaks from joints, flanges and pipelines must be attended at the earliest. Fuel oil should be free from possible contaminants such as dirt, sludge and water before it is fed to the combustion system.

LOSS OF EVEN ONE DROP OF OIL EVERY SECOND CAN COST YOU OVER 4000 LITRES A YEAR

Removal of Contaminants

Furnace oil arrives at the factory site either in tank lorries by road or by rail. Oil is then decanted into the main storage tank. To prevent contaminants such as rags, cotton waste, loose nuts or bolts or screws entering the system and damaging the pump, coarse strainer of 10 mesh size (not more than 3 holes per linear inch) is positioned on the entry pipe to the storage tanks.

Progressively finer strainers should be provided at various points in the oil supply system to filter away finer contaminants such as external dust and dirt, sludge or free carbon. It is advisable to provide these filters in duplicate to enable one filter to be cleaned while oil supply is maintained through the other. The Figure 1.1 gives an illustration of the duplex system of arrangement of strainers.

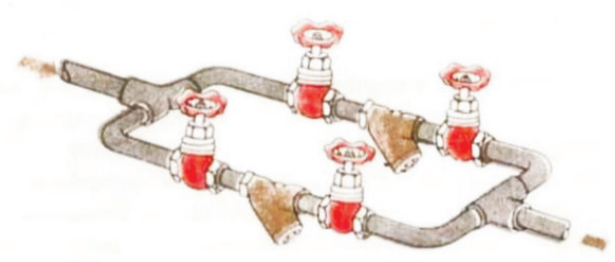


Figure 1.1 Duplex Arrangement of Strainers in a Pipeline

The Table 1.3 gives sizing of strainers at various locations.

Table 1.3 Sizing of Strainers

Location	Strainer Sizes	
	Mesh	Holes/Linear inch
Between rail/tank lorry decanting point and main storage tank	10	3
Between service tank and pre-heater	40	6
Between pre-heater and burner	100	10

Pumping

Heavy fuel oils are best pumped using positive displacement pumps, as they are able to get fuel moving when it is cold. A circulation gear pump running on LDO should give between 7000-10000 hours of service. Diaphragm pumps have a shorter service life, but are easier and less expensive to repair. A centrifugal pump is not recommended, because as the oil viscosity increases, the efficiency of the pump drops sharply and the horsepower required increases. Light fuels are best pumped with centrifugal or turbine pumps. When higher pressures are required, piston or diaphragm pumps should be used.

Fuel Storage and Pumping Temperature

The viscosity of furnace oil and LSHS increases with decrease in temperature, which makes it difficult to pump the oil. At low ambient temperatures (below 25 ^{o}C), furnace oil is not easily pump able. To circumvent this, preheating of oil is accomplished in two ways: a) the entire tank may be preheated. In this form of bulk heating, steam coils are placed at the bottom of the tank, which is fully insulated; b) the oil can be heated as it flows out with an outflow heater. To reduce steam requirements, it is advisable to insulate tanks where bulk heating is used.

Bulk heating may be necessary if flow rates are high enough to make outflow heaters of adequate capacity impractical, or when a fuel such as Low Sulphur Heavy Stock (LSHS) is used. In the case of outflow heating, only the oil, which leaves the tank, is heated to the pumping temperature. The outflow heater is essentially a heat exchanger with steam or electricity as the heating medium.

Temperature Control

Thermostatic temperature control of the oil is necessary to prevent overheating, especially when oil flow is reduced or stopped. This is particularly important for electric heaters, since oil may get carbonized when there is no flow and the heater is on. Thermostats should be provided at a region where the oil flows freely into the suction pipe. The temperature at which oil can readily be pumped depends on the grade of oil being handled. Oil should never be stored at a temperature above that necessary for pumping as this leads to higher energy consumption.

1.3 Properties of Coal

Coal Classification

Coal is classified into three major types namely anthracite, bituminous, and lignite. However there is no clear demarcation between them and coal is also further classified as semi- anthracite, semi-bituminous, and sub-bituminous. Anthracite is the oldest coal from geological perspective. It is a hard coal composed mainly of carbon with little volatile content and practically no moisture. Lignite is the youngest coal from geological perspective. It is a soft coal composed mainly of volatile matter and moisture content with low fixed carbon. Fixed carbon refers to carbon in its free state, not combined with other elements. Volatile matter refers to those combustible constituents of coal that vaporize when coal is heated.

The common coals used in Indian industry are bituminous and sub-bituminous coal. The gradation of Indian coal based on its calorific value is as follows:

Grade	Calorific Value Range (in kcal/kg)
A	Exceeding 6200
B	5600 – 6200
C	4940 – 5600
D	4200 – 4940
E	3360 – 4200
F	2400 – 3360
G	1300 – 2400

Normally D, E and F coal grades are available to Indian Industry.

The chemical composition of coal has a strong influence on its combustibility. The properties of coal are broadly classified as

1. Physical properties
2. Chemical properties

Physical Properties

Heating Value:

The heating value of coal varies from coal field to coal field. The typical GCVs for various coals are given in the Table 1.4.

Table 1.4 GCV for Various Coals

Parameter	Lignite (Dry Basis)	Indian Coal	Indonesian Coal	South African Coal
GCV (kcal/kg)	4,500*	4,000	5,500	6,000

* GCV of lignite on '*as received basis*' is 2500 –3000

Analysis of Coal

There are two methods: ultimate analysis and proximate analysis. The ultimate analysis determines all coal component elements, solid or gaseous and the proximate analysis determines only the fixed carbon, volatile matter, moisture and ash percentages. The ultimate analysis is determined in a properly equipped laboratory by a skilled chemist, while proximate analysis can be determined with a simple apparatus. It may be noted that proximate has no connection with the word "approximate".

Measurement of Moisture

Determination of moisture is carried out by placing a sample of powdered raw coal of size 200-micron size in an uncovered crucible and it is placed in the oven kept at 108 $\pm$ 2 ^{o}C along with the lid. Then the sample is cooled to room temperature and weighed again. The loss in weight represents moisture.

Measurement of Volatile Matter

Fresh sample of crushed coal is weighed, placed in a covered crucible, and heated in a furnace at 900 + 15 ^{o}C. For the methodologies including that for carbon and ash, refer to IS 1350 part I:1984, part III, IV. The sample is cooled and weighed. Loss of weight represents moisture and volatile matter. The remainder is coke (fixed carbon and ash).

Measurement of Carbon and Ash

The cover from the crucible used in the last test is removed and the crucible is heated over the Bunsen burner until all the carbon is burned. The residue is weighed, which is the incombustible ash. The difference in weight from the previous weighing is the fixed carbon. In actual practice, Fixed Carbon or FC derived by subtracting from 100 the value of moisture, volatile matter and ash.

Proximate Analysis

Proximate analysis indicates the percentage by weight of the Fixed Carbon, Volatiles, Ash, and Moisture Content in coal. The amounts of fixed carbon and volatile combustible matter directly contribute to the heating value of coal. Fixed carbon acts as a main heat generator during burning. High volatile matter content indicates easy ignition of fuel. The ash content is important in the design of the furnace grate, combustion volume, pollution control equipment and ash handling systems of a furnace. A typical proximate analysis of various coal is given in the Table 1.5.

Table 1.5 Typical Proximate Analysis of Various Coals (in Percentage)

Parameter	Indian Coal	Indonesian Coal	South African Coal
Moisture	5.98	9.43	8.5
Ash	38.63	13.99	17
Volatile matter	20.70	29.79	23.28
Fixed Carbon	34.69	46.79	51.22

Significance of Various Parameters in Proximate Analysis

a) Fixed carbon:

Fixed carbon is the solid fuel left in the furnace after volatile matter is distilled off. It consists mostly of carbon but also contains some hydrogen, oxygen, sulphur and nitrogen not driven off with the gases. Fixed carbon gives a rough estimate of heating value of coal.

b) Volatile Matter:

Volatile matters are the methane, hydrocarbons, hydrogen and carbon monoxide, and incombustible gases like carbon dioxide and nitrogen found in coal. Thus the volatile matter is an index of the gaseous fuels present. Typical range of volatile matter is 20 to 35%.

Volatile Matter

- Proportionately increases flame length, and helps in easier ignition of coal.
- Sets minimum limit on the furnace height and volume.
- Influences secondary air requirement and distribution aspects.
- Influences secondary oil support

c) Ash Content:

Ash is an impurity that will not burn. Typical range is 5 to 40%

Ash

- Reduces handling and burning capacity.
- Increases handling costs.
- Affects combustion efficiency and boiler efficiency
- Causes clinkering and slagging.

d) Moisture Content:

Moisture in coal must be transported, handled and stored. Since it replaces combustible matter, it decreases the heat content per kg of coal. Typical range is 0.5 to 10%

Moisture

- Increases heat loss, due to evaporation and superheating of vapour
- Helps, to a limit, in binding fines.
- Aids radiation heat transfer.

e) Sulphur Content:

Typical range is 0.5 to 0.8% normally.

Sulphur

- Affects clinkering and slagging tendencies
- Corrodes chimney and other equipment such as air heaters and economisers
- Limits exit flue gas temperature.

Chemical Properties

Ultimate Analysis:

The ultimate analysis indicates the various elemental chemical constituents such as Carbon, Hydrogen, Oxygen, Sulphur, etc. It is useful in determining the quantity of air required for combustion and the volume and composition of the combustion gases. This information is required for the calculation of flame temperature and the flue duct design etc. Typical ultimate analyses of various coals are given in the Table 1.6.

Table 1.6 Typical Ultimate Analyses of Coals

Parameter	Indian Coal, %	Indonesian Coal, %
Moisture	5.98	9.43
Mineral Matter (1.1 x Ash)	38.63	13.99
Carbon	41.11	58.96
Hydrogen	2.76	4.16
Nitrogen	1.22	1.02
Sulphur	0.41	0.56
Oxygen	9.89	11.88

Table 1.7 Relationship Between Ultimate Analysis and Proximate Analysis

	%C	=	0.97C+ 0.7(VM - 0.1A) - M(0.6-0.01M)
	%H	=	$0.036C + 0.086 (VM -0.1xA) - 0.0035M^2 (1-0.02M)$
	$\%N_2$	=	2.10 -0.020 VM
Where			
	C	=	% of fixed carbon
	A	=	% of ash
	VM	=	% of volatile matter
	M	=	% of moisture

Note: The above equation is valid for coal containing greater than 15% moisture content.

Storage, Handling and Preparation of Coal

Uncertainty in the availability and transportation of fuel necessitates storage and subsequent handling. Stocking of coal has its own disadvantages like build-up of inventory, space constraints, deterioration in quality and potential fire hazards. Other minor losses associated with the storage of coal include oxidation, wind and carpet loss. A 1% oxidation of coal has the same effect as 1% ash in coal, wind losses may account for nearly 0.5 – 1.0% of the total loss.

The main goal of good coal storage is to minimise carpet loss and the loss due to spontaneous combustion. Formation of a soft carpet, comprising of coal dust and soil causes carpet loss. On the other hand, gradual temperature builds up in a coal heap, on account of oxidation may lead to spontaneous combustion of coal in storage.

The measures that would help in reducing the carpet losses are as follows:

1. Preparing a hard ground for coal to be stacked upon.
2. Preparing standard storage bays out of concrete and brick

In process industry, modes of coal handling range from manual to conveyor systems. It would be advisable to minimise the handling of coal so that further generation of fines and segregation effects are reduced.

Preparation of Coal

Preparation of coal prior to feeding into the boiler is an important step for achieving good combustion. Large and irregular lumps of coal may cause the following problems:

1. Poor combustion conditions and inadequate furnace temperature.
2. Higher excess air resulting in higher stack loss.
3. Increase of unburnts in the ash.
4. Low thermal efficiency.

(a) Sizing of Coal

Proper coal sizing is one of the key measures to ensure efficient combustion. Proper coal sizing, with specific relevance to the type of firing system, helps towards even burning, reduced ash losses and better combustion efficiency.

Coal is reduced in size by crushing and pulverizing. Pre-crushed coal can be economical for smaller units, especially those which are stoker fired. In a coal handling system, crushing is limited to a top size of 6 or 4mm. The devices most commonly used for crushing are the rotary breaker, the roll crusher and the hammer mill.

It is necessary to screen the coal before crushing, so that only oversized coal is fed to the crusher. This helps to reduce power consumption in the crusher. Recommended practices in coal crushing are:

1. Incorporation of a screen to separate fines and small particles to avoid extra fine generation in crushing.
2. Incorporation of a magnetic separator to separate iron pieces in coal, which may damage the crusher.

The Table 1.8 gives the proper size of coal for various types of firing systems.

Table 1.8 Proper Size of Coal for Various Types of Firing System		
S. No.	**Types of Firing System**	**Size (in mm)**
1	Hand Firing (a) Natural draft (b) Forced draft	 25-75 25-40
2	Stoker Firing (a) Chain grate i) Natural draft ii) Forced draft (b) Spreader Stoker	 25-40 15-25 15-25
3	Pulverized Fuel Fired	75% below 75 micron*
4	Fluidized bed boiler	< 10 mm

*1 Micron = 1/1000 mm

(b) Conditioning of Coal

The fines in coal present problems in combustion on account of segregation effects. Segregation of fines from larger coal pieces can be reduced to a great extent by conditioning coal with water. Water helps fine particles to stick to the bigger lumps due to surface tension of the moisture, thus stopping fines from falling through grate bars or being carried away by the furnace draft. While tempering the coal, care should be taken to ensure that moisture addition is uniform and preferably done in a moving or falling stream of coal.

If the percentage of fines in the coal is very high, wetting of coal can decrease the percentage of unburnt carbon and the excess air level required to be supplied for combustion. Table 1.9 shows the extent of wetting, depending on the percentage of fines in coal.

Table 1.9 Extent of Wetting: Fines Vs Surface Moisture in Coal	
Fines (%)	**Surface Moisture (%)**
10 - 15	4 - 5
15 - 20	5 - 6
20 - 25	6 - 7
25 - 30	7 - 8

(c) Blending of Coal

In case of coal lots having excessive fines, it is advisable to blend the predominantly lumped coal with lots containing excessive fines. Coal blending may thus help to limit the extent of fines in coal being fired to not more than 25%. Blending of different qualities of coal may also help to supply a uniform coal feed to the boiler.

The proximate and ultimate analysis of various coal is given in Table 1.10 and 1.11.

Table 1.10 Proximate Analysis of Typical Coal				
	Lignite	**Bituminous Coal (Sample I)**	**Bituminous Coal (Sample II)**	**Indonesian Coal**
Moisture (%)	50	5.98	4.39	9.43
Ash (%)	10.41*	38.65	47.86	13.99
Volatile matter (%)	47.76*	20.70	17.97	29.79
Fixed carbon (%)	41.83*	34.69	29.78	46.79

* Dry Basis

Table 1.11 Ultimate Analysis of Various Coals			
	Bituminous Coal (Sample I)	**Bituminous Coal (Sample II)**	**Indonesian Coal**
Moisture (%)	5.98	4.39	9.43
Mineral matter (%)	38.63	47.86	13.99
Carbon (%)	42.11	36.22	58.96
Hydrogen (%)	2.76	2.64	4.16
Nitrogen (%)	1.22	1.09	1.02
Sulphur (%)	0.41	0.55	0.56
Oxygen (%)	9.89	7.25	11.88
GCV (kcal/kg)	4000	3500	5500

1.4 Properties of Gaseous Fuels

Gaseous fuels in common use are liquefied petroleum gases (LPG), Natural gas, producer gas, blast furnace gas, coke oven gas etc. The calorific value of gaseous fuel is expressed in kilo Calories per normal cubic meter (kcal/Nm^3) i.e. at normal temperature (20 ^oC) and pressure (760 mm Hg)

Calorific Value

Since most gas combustion appliances cannot utlilize the heat content of the water vapour, gross calorific value is of little interest. Fuel should be compared based on the net calorific value. This is especially true for natural gas, since increased hydrogen content results in high water formation during combustion.

Typical physical and chemical properties of various gaseous fuels are given in Table 1.12.

Table 1.12 Typical Physical and Chemical Properties of Various Gaseous Fuels					
Fuel Gas	**Relative Density**	**Higher Heating Value, kcal/Nm^3**	**Air/Fuel ratio, m^3 of air to m^3 of Fuel**	**Flame Temp., ^oC**	**Flame Speed, m/s**
Natural Gas	0.6	9350	10	1954	0.290
Propane	1.52	22200	25	1967	0.460
Butane	1.96	28500	32	1973	0.870

LPG

LPG is a predominant mixture of propane and Butane with a small percentage of unsaturates (Propylene and Butylene) and some lighter C_2 as well as heavier C_5 fractions. Included in the LPG range are propane (C_3H_8), Propylene (C_3H_6), normal and iso-butane (C_4H_{10}) and Butylene (C_4H_8).

LPG may be defined as those hydrocarbons, which are gaseous at normal atmospheric pressure, but may be condensed to the liquid state at normal temperature, by the application of moderate pressures. Although they are normally used as gases, they are stored and transported as liquids under pressure for convenience and ease of handling. Liquid LPG evaporates to produce about 250 times volume of gas.

LPG vapour is denser than air: butane is about twice as heavy as air and propane about one and half times as heavy as air. Consequently, the vapour may flow along the ground and into drains sinking to the lowest level of the surroundings and be ignited at a considerable distance from the source of leakage. In still air vapour will disperse slowly. Escape of even small quantities of the liquefied gas can give rise to large volumes of vapour / air mixture and thus cause considerable hazard. To aid in the detection of atmospheric leaks, all LPG's are required to be odorized. There should be adequate ground level ventilation where LPG is stored. For this very reason LPG cylinders should not be stored in cellars or basements, which have no ventilation at ground level.

Natural Gas

Methane is the main constituent of Natural gas and accounting for about 95% of the total volume. Other components are: Ethane, Propane, Butane, Pentane, Nitrogen, Carbon Dioxide, and traces of other gases. Very small amounts of sulphur compounds are also present. Since methane is the largest component of natural gas, generally properties of methane are used when comparing the properties of natural gas to other fuels.

Natural gas is a high calorific value fuel requiring no storage facilities. It mixes with air readily and does not produce smoke or soot. It has no sulphur content. It is lighter than air and disperses into air easily in case of leak. A typical comparison of carbon contents in oil, coal and gas is given in the Table 1.13.

Table 1.13 Comparison of Chemical Composition of Various Fuels

	Fuel Oil	Coal	Natural Gas
Carbon	84	41.11	74
Hydrogen	12	2.76	25
Sulphur	3	0.41	-
Oxygen	1	9.89	Trace
Nitrogen	Trace	1.22	0.75
Ash	Trace	38.63	-
Water	Trace	5.98	-

1.5 Properties of Agro Residues

The use of locally available agro residues is on the rise. This includes rice husk, coconut shells, groundnut shells, Coffee husk, Wheat stalk etc. The properties of a few of them are given in the Tables 1.14 and 1.15.

Table 1.14 Proximate Analysis of Typical Agro Residues

	Deoiled Bran	Paddy Husk	Saw Dust	Coconut Shell
Moisture	7.11	10.79	37.98	13.95
Ash	18.46	16.73	1.63	3.52
Volatile Matter	59.81	56.46	81.22	61.91
Fixed Carbon	14.62	16.02	17.15	20.62

Table 1.15 Ultimate Analysis of Typical Agro Residues

	Deoiled Bran	Paddy Husk	Saw Dust	Coconut Shell
Moisture	7.11	10.79	37.98	13.95
Mineral Matter	19.77	16.73	1.63	3.52
Carbon	36.59	33.95	48.55	44.95
Hydrogen	4.15	5.01	6.99	4.99
Nitrogen	0.82	0.91	0.80	0.56
Sulphur	0.54	0.09	0.10	0.08
Oxygen	31.02	32.52	41.93	31.94
GCV (kcal/kg)	3151	3568	4801	4565

Biomass Storage, Handling and Preparation

Biomass fuels have low bulk density which results in higher transportation cost. The transportation cost constitutes a significant portion of the landed cost of biomass. The low bulk density also requires vast area for storage. A common concern in biomass systems is the difficulty to ensure availability of any particular type of biomass throughout the entire year. A variety of types of biomass necessitate different types of collection and handling equipment. Most of the common biomass fuels, such as Woody biomass, Juliflora, Palm bunches, Jute Sticks, Cotton Stalks, De-oiled bran, Coir pith etc., require special types of handling machines, which add up to additional capital investment.

1.6 Combustion

Principle of Combustion

Combustion refers to the rapid oxidation of fuel accompanied by the production of heat, or heat and light. Complete combustion of a fuel is possible only in the presence of an adequate supply of oxygen.

Oxygen (O_2) is one of the most common elements on earth making up 20.9% of our air. Rapid fuel oxidation results in large amounts of heat. Solid or liquid fuels must be changed to a gas before they will burn. Usually heat is required to change liquids or solids into gases. Fuel gases will burn in their normal state if enough air is present.

Most of the 79% of air (that is not oxygen) is nitrogen, with traces of other elements. Nitrogen is considered to be a temperature reducing dilutant that must be present to obtain the oxygen required for combustion.

Nitrogen reduces combustion efficiency by absorbing heat from the combustion of fuels and diluting the flue gases. This reduces the heat available for transfer through the heat exchange surfaces. It also increases the volume of combustion by-products, which then have to travel through the heat exchanger and up the stack faster to allow the introduction of additional fuel air mixture.

This nitrogen also can combine with oxygen (particularly at high flame temperatures) to produce oxides of nitrogen (NO_x), which are toxic pollutants.

Carbon, hydrogen and sulphur in the fuel combine with oxygen in the air to form carbon dioxide, water vapour and sulphur dioxide, releasing 8084 kcal, 28922 kcal & 2224 kcal of heat respectively. Under certain conditions, Carbon may also combine with Oxygen to form Carbon Monoxide, which results in the release of a smaller quantity of heat (2430 kcal/kg of carbon) carbon burned to CO_2 will produce more heat per pound of fuel than when CO or smoke are produced.

C	**+ O_2**	**→**	**CO_2**	**+ 8084 kcal/kg of Carbon**
2C	**+ O_2**	**→**	**2 CO**	**+ 2430 kcal/kg of Carbon**
$2H_2$	**+ O_2**	**→**	**$2H_2O$**	**+ 28,922 kcal/kg of Hydrogen**
S	**+ O_2**	**→**	**SO_2**	**+ 2,224 kcal/kg of Sulphur**

Each kilogram of CO formed means a loss of 5654 kcal of heat (8084-2430).

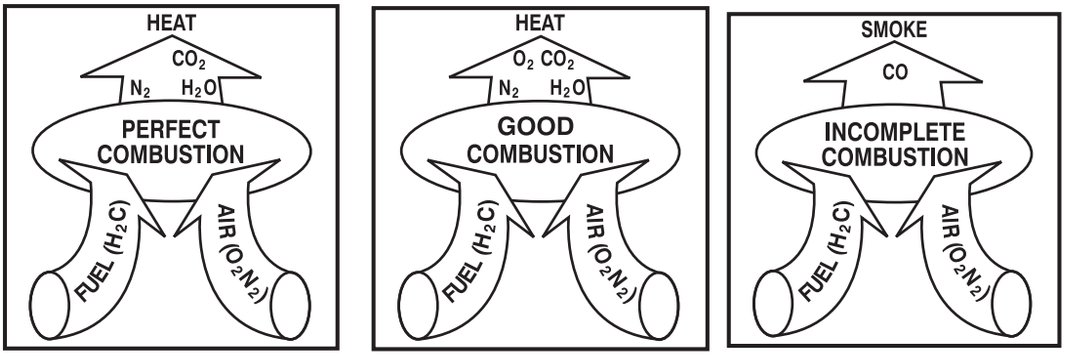
3 T's of Combustion

The objective of good combustion is to release all of the heat in the fuel. This is accomplished by controlling the "three T's" of combustion which are (1) Temperature high enough to ignite and maintain ignition of the fuel, (2) Turbulence or intimate mixing of the fuel and oxygen, and (3) Time sufficient for complete combustion.

Commonly used fuels like natural gas and propane generally consist of carbon and hydrogen. Water vapor is a by-product of burning hydrogen. This robs heat from the flue gases, which would otherwise be available for more heat transfer.

Natural gas contains more hydrogen and less carbon per kg than fuel oils and as such produces more water vapor. Consequently, more heat will be carried away by exhaust while firing natural gas.

Too much, or too little fuel with the available combustion air may potentially result in unburned fuel and carbon monoxide generation. A very specific amount of O_2 is needed for perfect combustion and some additional (excess) air is required for ensuring complete combustion. However, too much excess air will result in heat and efficiency losses.



Not all of the heat in the fuel are converted to heat and absorbed by the steam generation equipment. Usually all of the hydrogen in the fuel is burned and most boiler fuels, allowable with today's air pollution standards, contain little or no sulfur. So the main challenge in combustion efficiency is directed toward unburned carbon (in the ash or incompletely burned gas), which forms CO instead of CO_2.

1.7 Combustion of Oil

Heating Oil to Correct Viscosity

When atomizing oil, it is necessary to heat it enough to get the desired viscosity. This temperature varies slightly for each grade of oil. The lighter oils do not usually require pre-heating. Typical viscosity at the burner tip (for LAP, MAP & HAP burners) for furnace oil should be 100 Redwood seconds-1 which would require heating the oil to about 105 ^oC.

Stoichiometric Combustion

The efficiency of a boiler or furnace depends on efficiency of the combustion system. The amount of air required for complete combustion of the fuel depends on the elemental constituents of the fuel that is Carbon, Hydrogen, and Sulphur etc. This amount of air is called stoichiometric air. For ideal combustion process for burning one kg of a typical fuel oil containing 86% Carbon, 12% Hydrogen, 2% Sulphur, theoretically required quantity of air is 14.1 kg. This is the minimum air that would be required if mixing of fuel and air by the burner and combustion is perfect. The combustion products are primarily Carbon Dioxide (CO_2), water vapor (H_2 O) and Sulphur Dioxide (SO_2), which pass through the chimney along with the Nitrogen (N_2) in the air.

After surrendering useful heat in the heat absorption area of a furnace or boiler, the combustion products or fuel gases leave the system through the chimney, carrying away a significant quantity of heat with them.

Calculation of Stoichiometric Air

The specifications of furnace oil from lab analysis is given below:

Constituents	% By weight
Carbon	85.9
Hydrogen	12
Oxygen	0.7
Nitrogen	0.5
Sulphur	0.5
H_2O	0.35
Ash	0.05

GCV of fuel : 10880 kcal/kg

Calculation for Requirement of Theoretical Amount of Air

Considering a sample of 100 kg of furnace oil. The chemical reactions are:

Element	Molecular Weight kg / kg mole
C	12
O_2	32
H	2
S	32
N_2	28
CO_2	44
SO_2	64
H_2O	18

$C + O_2 \rightarrow CO_2$

$H_2 + ½\ O_2 \rightarrow H_2O$

$S + O_2 \rightarrow SO_2$

Constituents of fuel

$C + O_2 \rightarrow CO_2$

$12 + 32 \rightarrow 44$

12 kg of carbon requires 32 kg of oxygen to form 44 kg of carbon dioxide therefore 1 kg of carbon requires 32/12 kg i.e 2.67 kg of oxygen

$(85.9)\ C + (85.9 \times 2.67)\ O_2 \rightarrow 314.97\ CO_2$

$2H_2 + O_2 \rightarrow 2H_2O$
$4 + 32 \rightarrow 36$

4 kg of hydrogen requires 32 kg of oxygen to form 36 kg of water, therefore 1 kg of hydrogen requires 32/4 kg i.e 8 kg of oxygen

$(12)\ H_2 + (12 \times 8)\ O_2 \rightarrow (12 \times 9)\ H_2O$

$S + O_2 \rightarrow SO_2$
$32 + 32 \rightarrow 64$

32 kg of sulphur requires 32 kg of oxygen to form 64 kg of sulphur dioxide, therefore 1 kg of sulphur requires 32/32 kg i.e 1 kg of oxygen

$(0.5)\ S + (0.5 \times 1)\ O_2 \rightarrow 1.0\ SO_2$

Total Oxygen required (229.07+96+0.5) = 325.57 kg

Oxygen already present in 100 kg fuel (given) = 0.7 kg

Additional Oxygen Required = 325.57 – 0.7
= 324.87 kg

Therefore quantity of dry air reqd. (air contains 23% oxygen by wt.) = (324.87) / 0.23
= 1412.45 kg of air

Theoretical Air required = (1412.45) / 100
= 14.12 kg of air / kg of fuel

Calculation of theoretical CO_2 content in flue gases

Nitrogen in flue gas = 1412.45 – 324.87 + 0.5
= 1088.08 kg

Theoretical CO_2% in dry flue gas by volume is calculated as below :

Moles of CO_2 in flue gas = (314.97) / 44 = 7.16

Moles of N_2 in flue gas = (1088.08) / 28 = 38.86

Moles of SO_2 in flue gas = 1/64 = 0.016

$$Theoritical\ CO_2\%\ by\ volume = \frac{Moles\ of\ CO_2}{Total\ moles\ (dry)}\ x\ 100$$

$$= \frac{7.16}{7.16+38.86+0.016}\ x\ 100$$

$$= 15.6\ \%$$

Calculation of constituents of flue gas with excess air

% CO_2 measured in flue gas = 10% (measured)

$$\%\ \text{Excess air supplied (EA)} = \frac{7900\,x\,[(CO_2\%)_t - (CO_2\%)_a]}{(CO_2)_a\%\ x\ [100-(CO_2\%)_t]}$$

$$\%\ \text{Excess air supplied (EA)} = \frac{7900\,x\,[15.6-10]}{10\ x\ [100-15.6]}$$

$$= 52.4\%$$

Theoretical air required for 100 kg of fuel burnt = 1412.45 kg

Total quantity. of air supply required with 52.4% excess air = 1412.45 X 1.524

= 2152.57 kg

Excess air quantity = 2152.57 – 1412.45

= 740.12 kg.

O_2 = 740.12 X 0.23

= 170.23 kg

N_2 = 740.12 – 170.23

= 570 kg

The final constitution of flue gas with 52.4% excess air for every 100 kg fuel.

CO_2 = 314.97 kg

H_2O = 108.00 kg

SO_2 = 1 kg

O_2 = 170.23 kg

$$N_2 = 1088.08 + 570$$
$$= 1658.08 \text{ kg}$$

Determination of Actual CO_2% by calculation in dry flue gas by volume

Moles of CO_2 in flue gas = 314.97/44 = 7.16
Moles of SO_2 in flue gas = 1/64 = 0.016
Moles of O_2 in flue gas = 170.23 / 32 = 5.32
Moles of N_2 in flue gas = 1658.08 / 28 = 59.22

$$Actual\ CO_2\%\ by\ volume = \frac{Moles\ of\ CO_2}{Total\ moles\ (dry)}\ x\ 100$$

$$= \frac{7.16}{7.16 + 0.016 + 5.32 + 59.22}\ x\ 100$$

$$= \frac{7.16}{71.716}\ x\ 100 = 10\%$$

$$\text{Actual } O_2\% \text{ by volume} = \frac{5.32\,x100}{71.716}\ x\ 100 = 7.42\%$$

Optimizing Excess Air and Combustion

For complete combustion of every one kg of fuel oil 14.1 kg of air is needed. In practice, mixing is never perfect, a certain amount of excess air is needed to complete combustion and ensure that release of the entire heat contained in fuel oil. If too much air than what is required for completing combustion were allowed to enter, additional heat would be lost in heating the surplus air to the chimney temperature. This would result in increased stack losses. Less air would lead to the incomplete combustion and smoke. Hence, there is an optimum excess air level for each type of fuel.

Control of Air and Analysis of Flue Gas

Thus in actual practice, the amount of combustion air required will be much higher than optimally needed. Therefore some of the air gets heated in the furnace boiler and leaves through the stack without participating in the combustion

Chemical analysis of the gases is an objective method that helps in achieving finer air control. By measuring carbon dioxide (CO_2) or oxygen (O_2) in flue gases by continuous recording instruments or Orsat apparatus or portable fyrite, the excess air level as well as stack losses can be estimated with the graph as shown in Figure 1.2 and Figure 1.3. The excess air to be supplied depends on the type of fuel and the firing system. For optimum combustion of fuel oil, the CO_2 or O_2 in flue gases should be maintained at 14 -15% in case of CO_2 and 2-3% in case of O_2.

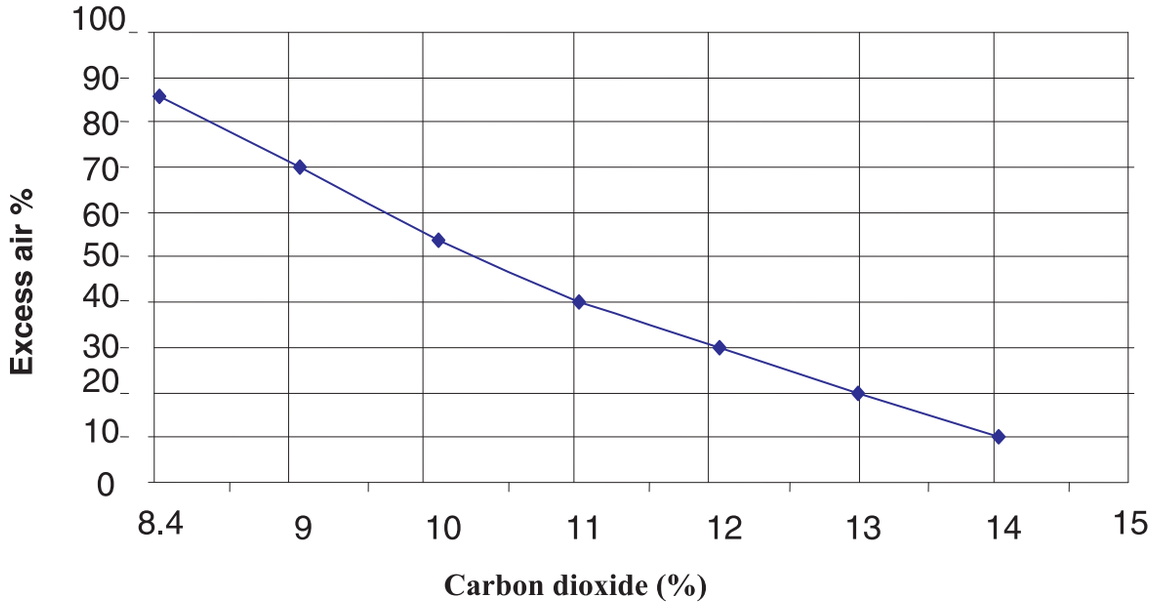


Figure 1.2 Relation Between CO_2 and Excess Air for Fuel Oil

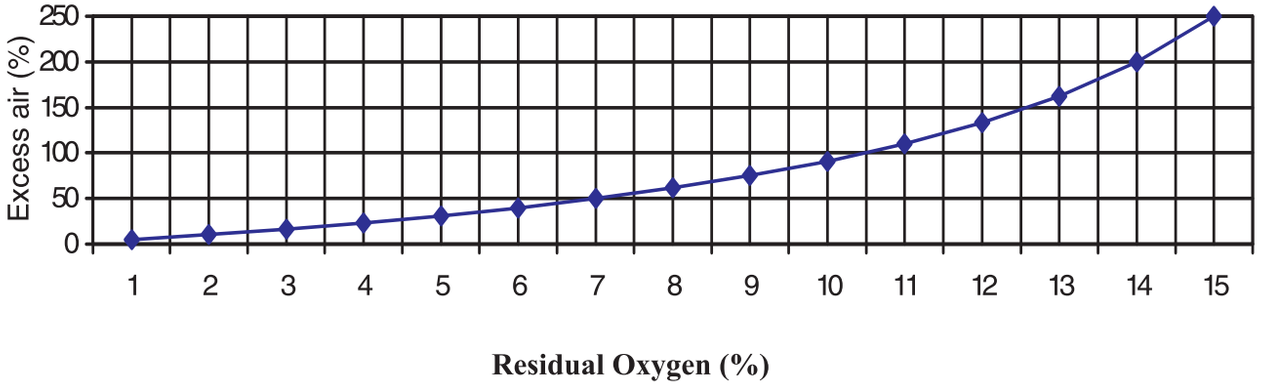


Figure 1.3 Relation Between Residual Oxygen and Excess Air

Oil Firing Burners

The burner is the principal device for the firing of fuel. The primary function of burner is to atomise fuel to millions of small droplets so that the surface area of the fuel is increased enabling intimate contact with oxygen in air. The finer the fuel droplets are atomised, more readily will the particles come in contact with the oxygen in the air and burn.

Normally, atomisation is carried out by primary air and completion of combustion is ensured by secondary air. Burners for fuel oil can be classified on the basis of the technique to prepare the fuel for burning i.e. atomisation.

Figure 1.4 shows a simplified burner head. The air is brought into the head by means of a forced draft blower or fan. The fuel is metered into the head through a series of valves. In order to get proper combustion, the air molecules must be thoroughly mixed with the fuel molecules before they actually burn. The air in the center is the primary air used for atomization and the one surrounding is the secondary air which ensures complete combustion.

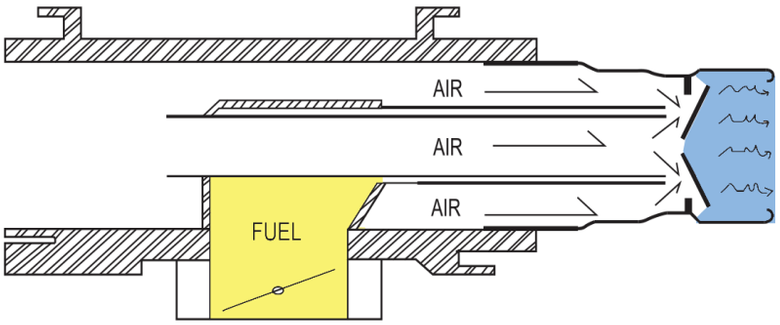


Figure 1.4 Burner Head

The mixing is achieved by burner parts designed to create high turbulence. If insufficient turbulence is produced by the burner, the combustion will be incomplete and samples taken at the stack will reveal carbon monoxide as evidence.

Since the velocity of air affects the turbulence, it becomes harder and harder to get good fuel and air mixing at higher turndown ratios since the air amount is reduced. Towards the highest turndown ratios of any burner, it becomes necessary to increase the excess air amounts to obtain enough turbulence to get proper mixing. The better burner design will be one that is able to properly mix the air and fuel at the lowest possible air flow or excess air.

An important aspect to be considered in selection of burner is **turndown ratio**. Turndown ratio is the relationship between the maximum and minimum fuel input without affecting the excess air level. For example, a burner whose maximum input is 250,000 kcal and minimum rate is 50,000 kcal, has a 'Turn-Down Ratio' of 5 to 1.

1.8 Combustion of Coal

Features of coal combustion

1 kg of coal will typically require 7-8 kg of air depending upon the carbon, hydrogen, nitrogen, oxygen and sulphur content for complete combustion. This air is also known as theoretical or stochiometric air.

If for any reason the air supplied is inadequate, the combustion will be incomplete. The result is poor

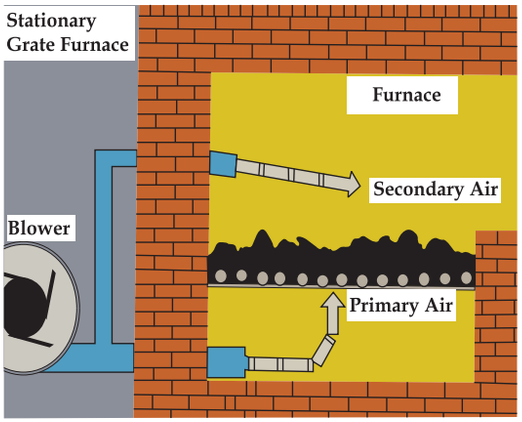


Figure 1.5 Coal Combustion

generation of heat with some portions of carbon remaining unburnt (black smoke) and forming carbon monoxide instead of carbon dioxides.

As in the case of oil, coal cannot be burnt with stochiometric quantity of air. Complete combustion is not achieved unless an excess of air is supplied.

The excess air required for coal combustion depends on the type of coal firing equipment. Hand fired boilers use large lumps of coal and hence need very high excess air. Stoker fired boilers as shown in the Figure 1.5 use sized coal and hence requires less excess air. Also in these systems primary air is supplied below the grate and secondary air is supplied over the grate to ensure complete combustion.

Fluidised bed combustion in which turbulence is created leads to intimate mixing of air and fuel resulting in further reduction of excess air. The pulverized fuel firing in which powdered coal is fired has the minimum excess air due to high surface area of coal ensuring complete combustion.

Clinker formation

Clinker is a mass of rough, hard, slag-like material formed during combustion of coal due to low fusion temperature of ash present in coal. Presence of silica, calcium oxide, magnesium oxides etc. in ash lead to a low fusion temperature. Typically Indian coals contain ash fusion temperature as low as 1100 ^{o}C. Once clinker is formed, it has a tendency to grow. Clinker will stick to a hot surface rather than a cold one and to a rough surface rather than a smooth one.

1.9 Combustion of Gas

Combustion Characteristics of Natural Gas

The stoichiometric ratio for natural gas (and most gaseous fuels) is normally indicated by volume. The air to natural gas (stoichiometric) ratio by volume for complete combustion vary between 9.5:1 to 10:1

Natural gas is essentially pure methane, CH_4. Its combustion can be represented as follows:

$$CH_4 + 2O_2 = CO_2 + 2H_2O$$

So for every 16 kgs of methane that are consumed, 44 kgs of carbon dioxide are produced. (Remember that the atomic weights of carbon, oxygen and hydrogen are 12, 16 and 1, respectively.)

Methane burns, when mixed with the proper amount of air and heated to the combustion temperature. Figure 1.6 shows the process with the amount of air and fuel required for perfect combustion.

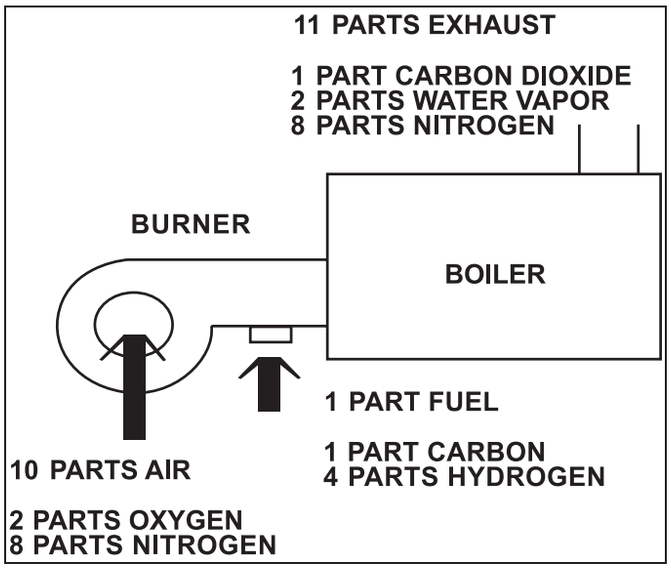


Figure 1.6 Combustion of Natural Gas

Low-And High-Pressure Gas Burners.

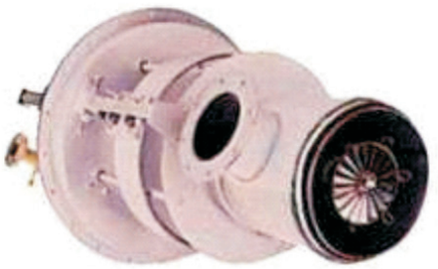


Figure 1.7 Low Pressure Gas Burner

The important thing in all gas-burning devices is a correct air-and-gas mixture at the burner tip. Low-pressure burners (Figure 1.7), using gas at a pressure less than 0.15 kg/cm^2 (2 psi), are usually of the multi-jet type, in which gas from a manifold is supplied to a number of small single jets, or circular rows of small jets, centered in or discharging around the inner circumference of circular air openings in a block of some heat-resisting material. The whole is encased in a rectangular cast-iron box, built into the boiler setting and having louver doors front to regulate the air supply. Draft may be natural, induced, or forced.

In a high-pressure gas mixer (figure 1.8), the energy of the gas jet draws air into the mixing chamber and delivers a correctly proportioned mixture to the burner. When the regulating valve is opened, gas flows through a small nozzle into a venturi tube (a tube with a contracted section). Entrainment of air with high-velocity gas in the narrow venturi section draws air in through large openings in the end. The gas-air mixture is piped to a burner. The gas-burner tip may be in a variety of forms. In a sealed-in tip type, the proper gas-air mixture is piped to the burner,

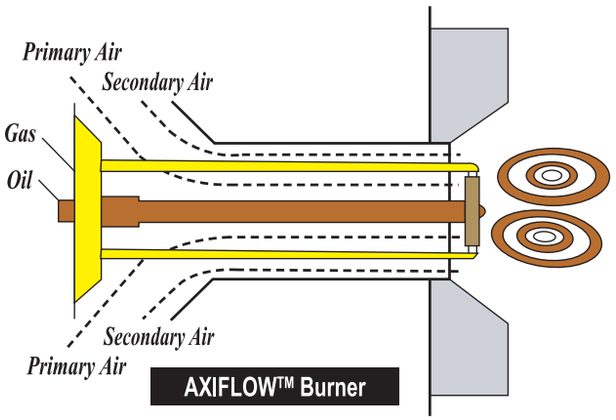


Figure 1.8 High Pressure Gas Mixer

and no additional air is drawn in around the burner tip. Size of the air openings in the venturi tube end is increased or decreased by turning a revolving shutter, which can be locked in any desired position. Excess air levels in natural gas burner are in the order of 5%.

1.10 Combustion of Biomass

Biomass can be converted into energy (heat or electricity) or energy carriers (charcoal, oil, or gas) using both thermochemical and biochemical conversion technologies. Combustion is the most developed and most frequently applied process used for solid biomass fuels because of its low costs and high reliability.

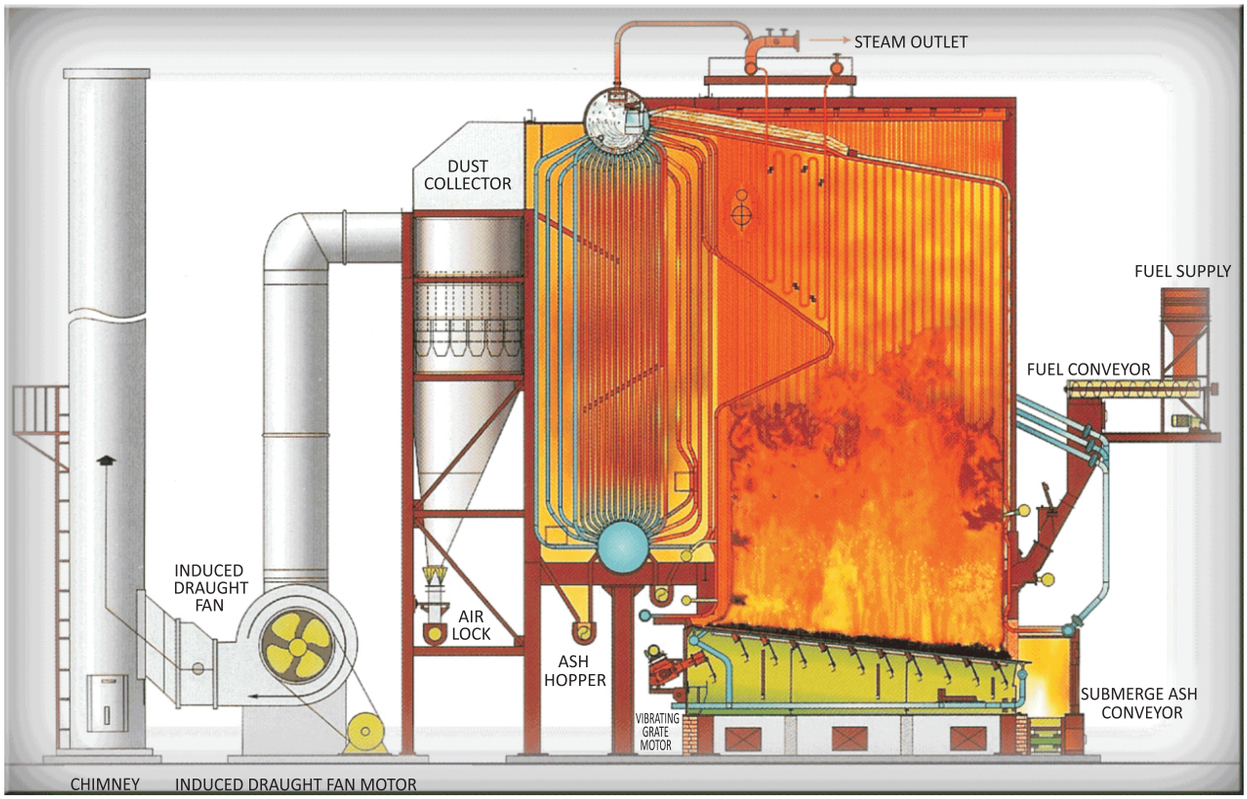


Figure 1.9 Biomass Combustion in a Boiler

During combustion, the biomass first loses its moisture at temperatures up to 100°C, using heat from other particles that release their heat value. As the dried particle heats up, volatile gases containing hydrocarbons, CO, CH4 and other gaseous components are released. In a combustion process, these gases contribute about 70% of the heating value of the biomass. Finally, char oxidises and ash remains. The combustion installation needs to be properly designed for a specific fuel type in order to guarantee adequate combustion quality and low emissions.

The characteristics and quality of biomass as a fuel depend on the kind of biomass and the pre-treatment technologies applied. For example, the moisture content of the fuel as fed into the furnace may vary from 25 - 55% (on a wet weight basis) for bark and sawmill by-products, and be less than 10% (on a

wet weight basis) for pellets. Also, the ash sintering temperatures of biofuels used cover a wide range (800 to 1200°C), as do particle shapes and sizes.

In order to reduce its moisture content, freshly harvested wood is often left outside for a number of weeks before it is chipped and fed to a combustion plant.

Different biomass combustion systems are available for industrial purposes. Broadly, they can be defined as fixed-bed combustion, fluidised bed combustion, and dust combustion.

Fixed-bed combustion:

Fixed-bed combustion systems include grate furnaces and underfeed stokers. Primary air passes through a fixed bed, where drying, gasification, and charcoal combustion take place in consecutive stages. The combustible gases are burned in a separate combustion zone using secondary air. Grate furnaces are appropriate for burning biomass fuels with high moisture content, different particle sizes, and high ash content. The design and control of the grate are aimed at guaranteeing smooth transportation and even distribution of the fuel and a homogeneous primary air supply over the whole grate surface. Irregular air supply may cause slagging, and higher amounts of fly ash, and may increase the oxygen needed for complete combustion. Load changes can be achieved more easily and quickly than in grate furnaces because there is better control of the fuel supply.

Fluidised bed combustion

In a fluidised bed, biomass fuel is burned in a self-mixing suspension of gas and solid bed material (usually silica sand and dolomite) in which air for combustion enters from below. Depending on the fluidisation velocity, bubbling and circulating fluidised bed combustion can be distinguished.

The intense heat transfer and mixing provide good conditions for complete combustion with low excess air demand. The low excess air amounts required reduce the flue gas volume flow and increase combustion efficiency. Fluid bed combustion plants are of special interest for largescale applications (normally exceeding 30 MWth). For smaller plants, fixed bed systems are usually more cost-effective. One disadvantage is the high dust loads in the flue gas, which make efficient dust precipitators and boiler cleaning systems necessary. Bed material is also lost with the ash, making it necessary to periodically add new bed material.

Dust combustion

Dust combustion is suitable for fuels available as small, dry particles such as wood dust. A mixture of fuel and primary combustion air is injected into the combustion chamber. Combustion takes place while the fuel is in suspension; the transportation air is used as primary air. Gas burnout is achieved after secondary air addition. An auxiliary burner is used to start the furnace. When the combustion temperature reaches a certain value, biomass injection starts and the auxiliary burner is shut down. Due to the explosion-like gasification process of the biomass particles, careful fuel feeding is essential. Fuel gasification and charcoal combustion take place at the same time because of the small particle

size. Therefore, quick load changes and efficient load control can be achieved. Since the fuel and air are well-mixed, only a small amount of excess air is required. This results in high combustion efficiencies.

1.11 Draft System

The function of draft in a combustion system is to exhaust the products of combustion into the atmosphere. The draft systems can be classified into two types namely Natural and Mechanical Draft.

Natural Draft : It is the draft produced by a chimney alone. It is caused by the difference in weight between the column of hot gas inside the chimney and column of outside air of the same height and cross section. Being much lighter than outside air, chimney flue gas tends to rise, and the heavier outside air flows in through the ash pit to take its place. It is usually controlled by hand-operated dampers in the chimney and breeching connecting the boiler to the chimney. Here no fans or blowers are used. The products of combustion are discharged at such a height that it will not be a nuisance to the surrounding community.

Mechanical Draft : It is the draft artificially produced by fans. Three basic types of draft systems available are:

Balanced Draft : Forced-draft (F-D) fan (blower) pushes air into the furnace and an induced-draft (I-D) fan draws gases into the chimney thereby providing draft to remove the gases from the boiler. Here the pressure is maintained between 0.05 to 0.10 in. of water gauge below atmospheric pressure in the case of boilers and slightly positive for reheating and heat treatment furnaces.

Induced Draft : An induced-draft fan draws enough draft for flow into the furnace, causing the products of combustion to discharge to atmosphere. Here the furnace is kept at a slight negative pressure below the atmospheric pressure so that combustion air flows through the system.

Forced Draft : The Forced draft system uses a fan to deliver the air to the furnace, forcing combustion products to flow through the unit and up the stack.

1.12 Combustion Controls

Combustion controls assist the burner in regulation of fuel supply, air supply, (fuel to air ratio), and removal of gases of combustion to achieve optimum boiler efficiency. The amount of fuel supplied to the burner must be in proportion to the steam pressure and the quantity of steam required. The combustion controls are also necessary as safety device to ensure that the boiler operates safely.

Various types of combustion controls in use are:

ON/OFF Control

The simplest control, ON/OFF control means that either the burner is firing at full rate or it is OFF. This type of control is limited to small boilers.

High/Low/OFF Control

Slightly more complex is HIGH/LOW/OFF system where the burner has two firing rates. The burner operates at slower firing rate and then switches to full firing as needed. Burner can also revert to low firing position at reduced load. This control is fitted to medium sized boilers.

Modulating Control

The modulating control operates on the principle of matching the steam pressure demand by altering the firing rate over the entire operating range of the boiler. Modulating motors use conventional mechanical linkage or electric valves to regulate the primary air, secondary air, and fuel supplied to the burner. Full modulation means that boiler keeps firing, and fuel and air are carefully matched over the whole firing range to maximize thermal efficiency.

<u>Solved Example</u>:

For combustion of 500 lit/hr of furnace oil, estimate combustion air quantity per hour with 20% excess air. Specific gravity of furnace oil 0.95. (Fuel analysis: C - 84%, H_2 -12%, S - 3%, O_2 - 1%).

Ans:

$C + O_2 \longrightarrow CO_2$ (0.84 * 32/12) = 2.24 kg of O_2/kg of C
$12 + 32 \longrightarrow 44$

$H_2 + ½\ O_2 \longrightarrow H_2O$ (0.12 x 16/2) = 0.96 kg of O_2/ kg of H_2
$2 + 16 \longrightarrow 18$

$S + O_2 \longrightarrow SO_2$ (0.03 *32/32) = 0.03 kg of O_2 / kg of S
$32 + 32 \longrightarrow 64$

Total O_2 required = (2.24 + 0.96 + 0.03 – 0.01)
= 3.22 kg of O_2 / kg furnace oil

Theoretical Air required = 3.22 / 0.23 = 14 kg of air / kg furnace oil

Excess air = 20%

Air supplied = (1 + 0.2) x 14 = 16.8 kg of air / kg of oil

Quantity of furnace oil = 500 lit/hr. = (500 x 0.95) = 475 kg/hr

Total quantity of air required = 475 x 16.8 = 7980 kgs of air / hr.

QUESTIONS	
Objective Type Questions	
1.	Which fuel requires the lowest amount of excess air for combustion? a) furnace oil b) LDO c) coal d) natural gas
2.	Which fuel requires maximum air for stoichiometric combustion? a) butane b) propane c) hydrogen d) coal
3.	How many kg of CO_2 are produced in complete combustion of 16 kg of methane? a) 42 b) 44 c) 16 d) none of the above
4.	Which agro-residue has the lowest gross calorific value? a) deoiled Bran b) paddy husk c) saw dust d) coconut shell
5.	Which among the following releases the most energy per kg on complete combustion? a) carbon b) sulphur c) nitrogen d) hydrogen
6.	Which among the following is most viscous fluid? a) furnace oil b) HSD c) kerosene d) water
7.	The content of solid left in the oven after volatile matter is distilled off, while analysing the composition of coal in a laboratory is _____. a) only sulphur b) only moisture c) only ash d) mix of fixed carbon and ash
8.	Which is the best suited pump for pumping LSHS? a) vertical turbine pump b) centrifugal pump c) diaphragm pump d) gear pump
9.	Oxygen percentage (by volume) measurement in flue gas can be done by using: a) ultrasonic tester b) potassium oxide probe c) copper tubes d) zirconium oxide probe
10.	For complete combustion of every kg of FO firing, the approximate theoretical quantity of air required is: a) 14 kg b) 18 kg c) 21 kg d) 10 kg
Short Type Questions	
S-1	Explain why natural gas requires least amount of excess air?
S-2	What is volatile matter and what is its significance?
S-3	The measured CO_2 is 8% in an oil fired boiler flue gas. Theoretical CO_2 content for the fuel fired is 16%. Estimate the %excess air level?
S-4	One kg of wood contains 20% moisture and 5% hydrogen by weight. How much water is evaporated during complete combustion of one kg of wood?

S-5	What is the difference between Gross calorific value (GCV) and Net calorific value (NCV)?
Long Type Questions	
L-1	What is draft in combustion system? Explain briefly the different types of draft systems
L-2	i) State the stoichiometric combustion equation for Methane. ii) How many kg of carbon dioxide is generated by complete combustion of 1 kg of Methane? iii) How many kg of water is generated by complete combustion of 1 kg of Methane?

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3. www.pcra.org

Chapter 2

Boilers

Learning Objectives

In this chapter you will learn about

- Heat transfer modes in boiler
- IBR definition of boiler and steam pipes
- Boiler system components, types and classifications
- Performance evaluation: direct and indirect method
- Boiler blow down concepts
- Boiler water treatment and importance
- Thermic fluid heaters
- Energy efficiency opportunities in boilers

2. BOILERS

Types, Combustion, Performance Evaluation, Feed Water Treatment, Blow Down, Improving Availability (Reducing Tube Failure, Soot Blowing and Reducing Soot Deposition, Preservation, Start-up and Shutdown procedure), Thermic Fluid Heaters, Energy Conservation Opportunities.

2.1 Introduction

A boiler is an enclosed vessel that provides a means for combustion heat to be transferred into water until it becomes heated water or steam. The hot water or steam under pressure is then usable for transferring the heat to a process. Water is a useful and cheap medium for transferring heat to a process. When water is boiled into steam its volume increases about 1,600 times, producing a force that is almost as explosive as gunpowder. This causes the boiler to be extremely dangerous equipment that must be treated with utmost care.

The process of heating a liquid until it reaches its gaseous state is called evaporation. Heat is transferred from one body to another by means of (1) radiation, which is the transfer of heat from a hot body to a cold body without a conveying medium, (2) convection, the transfer of heat by a conveying medium, such as air or water and (3) conduction, transfer of heat by actual physical contact, molecule to molecule.

Boiler Specification

The heating surface is any part of the boiler metal that has hot gases of combustion on one side and water on the other. Any part of the boiler metal that actually contributes to making steam is heating surface. The amount of heating surface of a boiler is expressed in square meters and can be related to the steam raising capacity of the boiler. The quantity of the steam produced is indicated in tons of water, evaporated to steam per hour. Maximum continuous rating is the hourly evaporation that can be maintained for 24 hours. F & A means the amount of steam generated from water at 100°C to saturated steam at 100 °C

Typical Boiler Specification

Boiler Make & Year	: XYZ & 2003
MCR (Maximum Continuous Rating): 10 TPH (F & A 100°C)	
Rated Working Pressure	:10.54 kg/cm^2(g)
Type of Boiler	: 3 Pass Fire tube
Fuel Fired	: Fuel Oil

Indian Boiler Regulation

The Indian Boilers Act was enacted to consolidate and amend the law relating to steam boilers. Indian Boilers Regulation (IBR) was created in exercise of the powers conferred by section 28 & 29 of the Indian Boilers Act **and further amended by the Indian Boilers (Amendment) Act, 2007**

Boilers means a pressure vessel in which steam is generated for use external to itself by application of heat which is wholly or partly under pressure when steam is shut off but does not include a pressure vessel.

i) With capacity less than 25 litres (such capacity being measured from the feed check valve to the main steam stop valve

ii) With less than one kilogram per centimeter square design gauge pressure and working gauge or

iii) In which water is heated below one hundred degrees centigrade

"Boiler component" means steam piping, feed piping, economizer, superheater, any mounting or other fitting and any other external or internal part of a boiler which is subject to pressure exceeding one kilogram per centimeter square gauge.

"Steam pipe" means any pipe through which steam passes if

i) The pressure at which steam passes through such pipe exceeds 3.5 kilogram per square centimeters above atmospheric pressure, or

ii) Such pipe exceeds 254 millimeters in internal diameter and the pressure of steam exceeds 1 kilogram per square centimeters above the atmospheric pressure and includes in either case any connected fitting of a steam pipe.

2.2 Boiler Systems

The boiler system comprises of feed water system, steam system and fuel system. The **feed water system** provides water to the boiler and regulates it automatically to meet the steam demand. Various valves provide access for maintenance and repair. The **steam system** collects and controls the steam produced in the boiler. Steam is directed through a piping system to the point of use. Throughout the system, steam pressure is regulated using valves and checked with steam pressure gauges. The **fuel system** includes all equipment used to provide fuel to generate the necessary heat. The equipment required in the fuel system depends on the type of fuel used in the system. A typical boiler system schematic is shown in Figure 2.1.

In the figure **A.** Flue gas heat recovery system using economizer **B.** Heat recovery from continuous blow down **C.** Utilisation of high pressure steam in the process **D.** use of low pressure steam (reduced by PRV) in the process **E.** Condensate return pump.

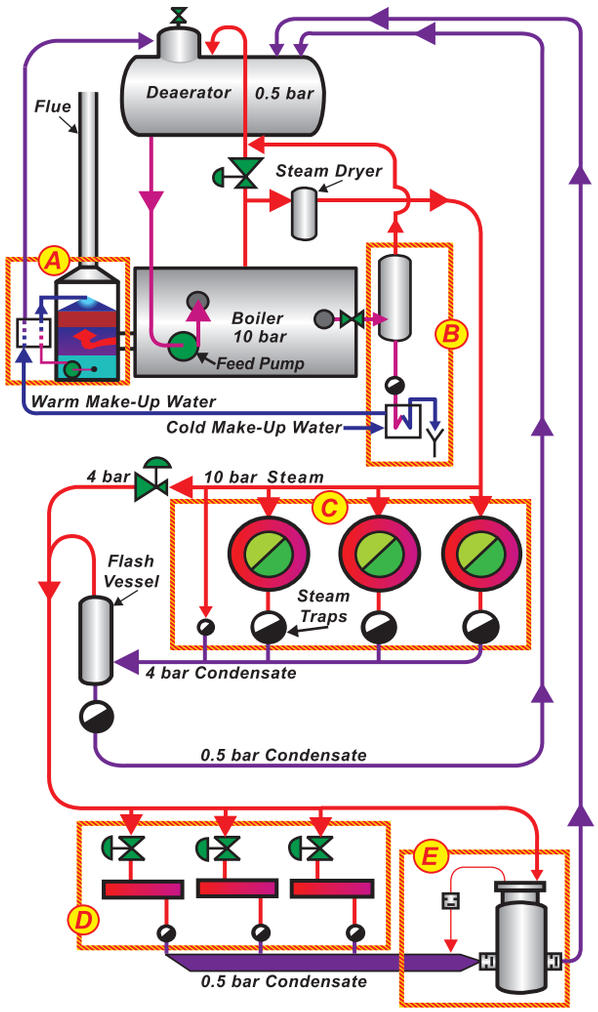


Figure 2.1 Boiler System Schematic

The water supplied to the boiler that is converted into steam is called **feed water**. The two sources of feed water are: (1) **Condensate** or condensed steam returned from the processes and (2) **Makeup water** (treated raw water) which must come from outside the boiler room and plant processes. In high capacity boilers, for higher efficiencies, besides the economizer for preheating the water an air preheater is also installed to preheat the combustion air.

2.3 Boiler Types and Classifications

There are virtually infinite numbers of boiler designs but generally they fit into one of the two categories:

Fire tube or "fire in tube" boilers; contain long steel tubes through which the hot gasses from a furnace pass and around which the water to be converted to steam circulates. (Refer Figure 2.2). Fire tube boilers, typically have a lower initial cost, are more fuel efficient and easier to operate, but they are limited generally to capacities of 25 tons/hr and pressures of 17.5 kg/cm^2.

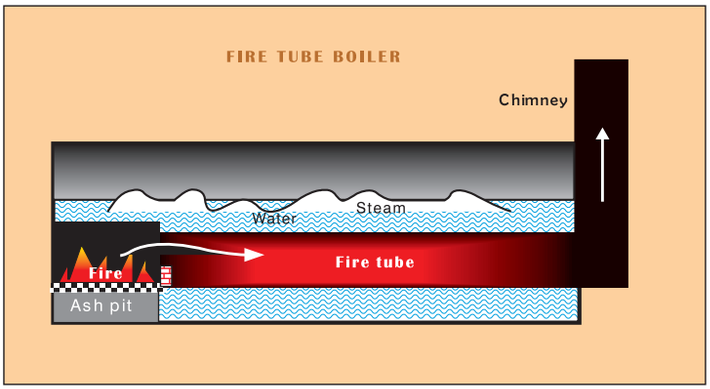


Figure 2.2 Fire Tube Boiler

Water tube or "water in tube" boilers in which the conditions are reversed with the water passing through the tubes and the hot gasses passing outside the tubes (see Figure 2.3). These boilers can be of single- or multiple-drum type. These boilers can be built to any steam capacities and pressures, and have higher efficiencies than fire tube boilers.

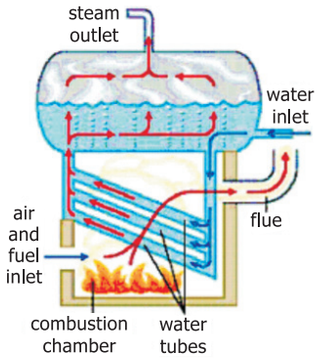


Figure 2.3 Water Tube Boiler

Packaged Boiler: The packaged boiler is so called because it comes as a complete package. Once delivered to site, it requires only the steam, water pipe work, fuel supply and electrical connections to be made for it to become operational. Package boilers are generally of shell type with fire tube design so as to achieve high heat transfer rates by both radiation and convection (Refer Figure 2.4).

The features of package boilers are:

- Small combustion space and high heat release rate resulting in faster evaporation.
- Large number of small diameter tubes leading to good convective heat transfer.
- Forced or induced draft systems resulting in good combustion efficiency.
- Number of passes resulting in better overall heat transfer.
- Higher thermal efficiency levels compared with other boilers.

These boilers are classified based on the number of passes - the number of times the hot combustion gases pass through the boiler. The combustion chamber is taken, as the first pass after which there may be one, two or three sets of fire-tubes. The most common boiler of this class is a three-pass unit with two sets of fire-tubes and with the exhaust gases exiting through the rear of the boiler.

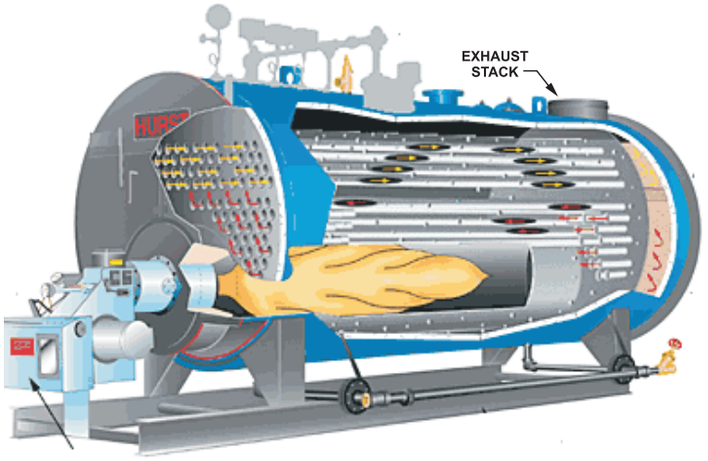


Figure 2.4 Packaged Boiler

Stoker Fired Boiler:

Stokers are classified according to the method of feeding fuel to the furnace and by the type of grate. The main classifications are:

1. Chain-grate or travelling-grate stoker
2. Spreader stoker

Chain-Grate or Travelling-Grate Stoker Boiler

Coal is fed at one end of a moving steel chain grate. As grate moves along the length of the furnace, the coal burns before dropping off at the end as ash. Some degree of skill is required, particularly when setting up the grate, air dampers and baffles, to ensure clean combustion leaving minimum of unburnt carbon in the ash.

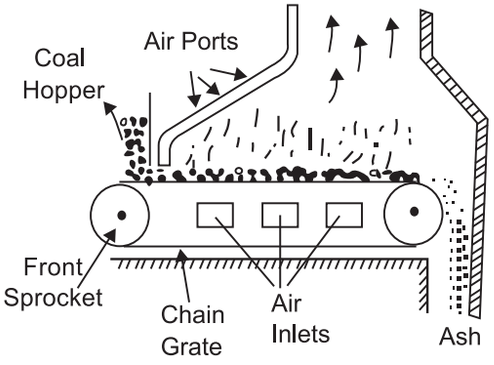


Figure 2.5 Chain Grate Stoker

The coal-feed hopper runs along the entire coal-feed end of the furnace. A coal grate is used to control the rate at which coal is fed into the furnace, and to control the thickness of the coal bed and speed of the grate. Coal must be uniform in size, as large lumps will not burn out completely by the time they reach the end of the grate. As the bed thickness decreases from coal-feed end to rear end, different amounts of air are required- more quantity at coal-feed end and less at rear end (see Figure 2.5).

Spreader Stoker Boiler

Spreader stokers utilize a combination of suspension burning and grate burning. The coal is continually fed into the furnace above a burning bed of coal. The coal fines are burned in suspension; the larger particles fall to the grate, where they are burned in a thin, fast-burning coal bed. This method of firing provides good flexibility to meet load fluctuations, since ignition is almost instantaneous when firing rate is increased. Hence, the spreader stoker is favored over other types of stokers in many industrial applications.

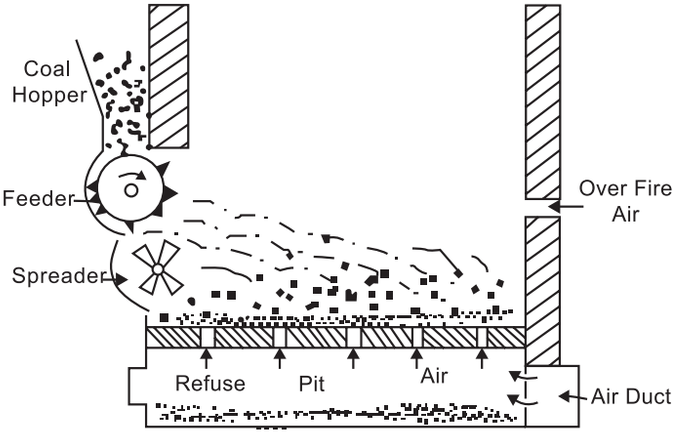


Figure 2.6 Spreader Stoker

Pulverized Fuel Boiler

Most coal-fired power station boilers use pulverized coal, and many of the larger industrial water-tube boilers also use this pulverized fuel. This technology is well developed, and there are thousands of units around the world, accounting for well over 90% of coal-fired capacity.

The coal is ground (pulverized) to a fine powder, so that less than 2% is +300 micro metre (µm) and 70-75% is below 75 microns, for a bituminous coal. It should be noted that too fine a powder is wasteful of grinding mill power. On the other hand, too coarse a powder does not burn completely in the combustion chamber and results in higher unburnt losses.

The pulverized coal is blown with part of the combustion air into the boiler plant through a series of burner nozzles. Secondary and tertiary air may also be added. Combustion takes place at temperatures from 1300-1700°C, depending largely on coal grade. Particle residence time in the boiler is typically 2 to 5 seconds, and the particles must be small enough for complete combustion to have taken place during this time.

This system has many advantages such as ability to fire varying quality of coal, quick responses to changes in load, use of high pre-heat air temperatures etc.

One of the most popular systems for firing pulverized coal is the tangential firing using four burners corner to corner to create a fireball at the center of the furnace (see Figure 2.7).

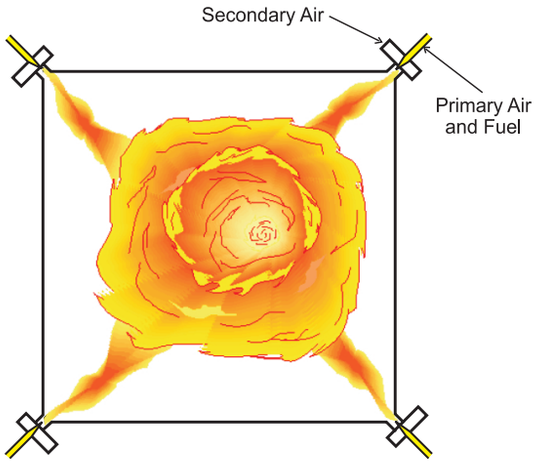


Figure 2.7 Tangential Firing

FBC Boiler

When an evenly distributed air or gas is passed upward through a finely divided bed of solid particles such as sand supported on a fine mesh, the particles are undisturbed at low velocity. As air velocity is gradually increased, a stage is reached when the individual particles are suspended in the air stream. Further, increase in velocity gives rise to bubble formation, vigorous turbulence and rapid mixing and the bed is said to be fluidized (Figure 2.8).

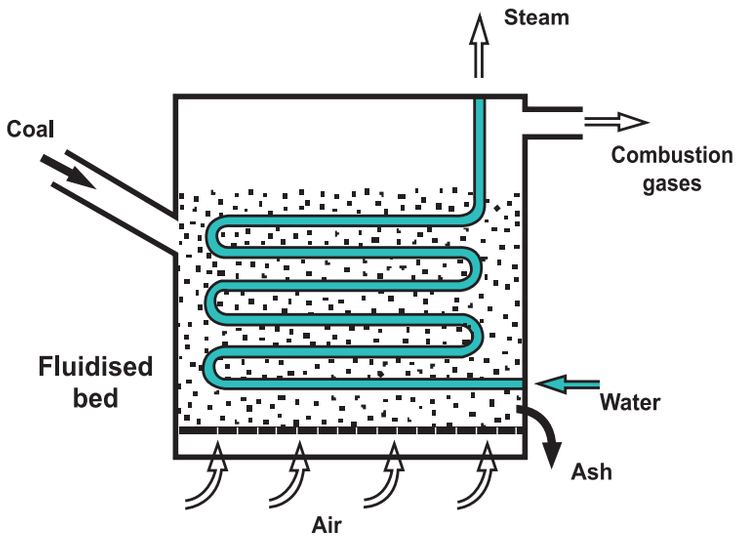


Figure 2.8 Fluidised Bed Combustion

If the sand in a fluidized state is heated to the ignition temperature of the coal and the coal is injected continuously in to the bed, the coal will burn rapidly, and the bed attains a uniform temperature due to effective mixing. Proper air distribution is vital for maintaining uniform fluidisation across the bed.). Ash is disposed by dry and wet ash disposal systems.

Fluidised bed combustion has significant advantages over conventional firing systems and offers multiple benefits namely fuel flexibility, reduced emission of noxious pollutants such as SOx and NOx, compact boiler design and higher combustion efficiency. More details about FBC boilers are given in Chapter 6 on Fluidized Bed Boiler.

Super Critical Boiler

In the temperature entropy diagram of steam, a point is reached where the boiling water and dry saturated steam lines converge and at that point, the latent heat is zero. The critical point corresponds to a pressure of 221.2 bar absolute and a temperature of 374.18 °C.

If water is heated beyond the above condition, steam parameters are referred to as super critical. A boiler producing steam above the critical pressure is called the supercritical boiler. While sub-critical boiler has three distinct sections - economiser, evaporator and superheater the supercritical boiler has only an economiser and superheater. The advantages of super critical boilers are

- Higher heat transfer rate
- More flexible in accepting load variation
- Grater ease of operation
- High thermal efficiency (40-42% of power generating stations)
- The absence of two-phase mixer minimise the problems of erosion and corrosion
- Steadier pressure level

Typical improvements in station heat rate with increasing pressures and temperatures are given in Table 2.1.

Table 2.1 Steam conditions and % improvement in heat rate	
Steam Parameters at turbine inlet (bar)/ (Superheat)°C/ (Reheat)°C)	**% improvement in station heat rate**
170 /538/538	Base case
170/ 538/565	0.5
170/ 565/565	1.3
246/538/538	1.6
246/538/565	2.1
246/565/565	3
246/565/598	3.6
306/598/598	5

*Source: BHEL

The super critical boilers call for special materials to be used for constituent heat transfer surfaces like drum, water walls, economizer and re-heaters, in order to withstand the elevated pressure & temperature conditions.

2.4 Performance Evaluation of Boilers

The performance parameters of a boiler like evaporation ratio and efficiency deteriorate with time due to poor combustion, heat transfer surface fouling and poor operation and maintenance. Even for a new boiler, reasons such as deteriorating fuel quality, water quality etc, can result in poor boiler performance. Evaporation ratio trending and Boiler efficiency tests help to find out the deviation of boiler efficiency from the rated values and target problem areas for corrective action.

Boiler Evaporation Ratio

Boiler Evaporation ratio, also known as Steam to fuel ratio, is a common and simple indicator for boiler performance trending.

Evaporation ratio monitoring is best suited for any boiler when its own performance is compared on day to day basis as a performance indicator; given that enthalpy gain in steam and fuel calorific value remain constant. A drop in evaporation ratio indicates a drop in Boiler efficiency.

Typical values of Evaporation ratio for different type of fuels are as follows:

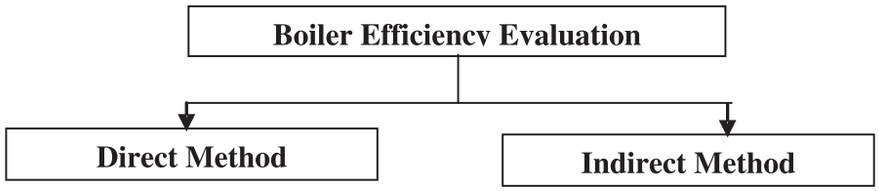
Biomass fired boilers : 2.0 to 3.0
Coal fired boilers : 4.0 to 5.5
Oil fired boilers : 13.5 to 14.5
Gas fired boilers : 11.0 to 13.0

Boiler Efficiency

Thermal efficiency of a boiler is defined as the percentage of heat input that is effectively utilized to generate steam. There are two methods of assessing boiler efficiency.

a) Direct method: Where the energy gain of the working fluid (water and steam) is compared with the energy content of the boiler fuel.

b) Indirect or Loss method: Where the efficiency is the difference between the losses and the energy input



a. Direct Method

This is also known as 'input-output method' due to the fact that it needs only the useful output (steam) and the heat input (i.e. fuel) for evaluating the efficiency. This efficiency can be evaluated using the formula

$$Boiler\ Efficiency = \frac{Heat\ Output}{Heat\ Input}\ x\ 100$$

Parameters to be monitored for the calculation of boiler efficiency by direct method are :

- Quantity of steam generated per hour (Q) in kg/hr.
- Quantity of fuel used per hour (q) in kg/hr.
- The working pressure (in kg/cm^2(g)) and superheat temperature (^{o}C), if any
- The temperature of feed water (^{o}C)
- Type of fuel and gross calorific value of the fuel (GCV) in kcal/kg of fuel

$$Boiler\ Efficiency\ (\eta) = \frac{Q\,x\,(h_g - h_f)}{q\,x\,GCV}\ x\ 100$$

Where, h_g – Enthalpy of saturated steam in kcal/kg of steam
h_f - Enthalpy of feed water in kcal/kg of water

The fuel calorific value may be gross or net and accordingly, the efficiency reported is referred to as efficiency on GCV or NCV basis.

Example

Find out the efficiency and evaporation ratio of the boiler by direct method with the data given below:

Type of boiler	: Coal fired
Quantity of steam (dry) generated	: 8 TPH
Steam pressure / temp	: 10 kg/cm^2(g)/ 180 ^{0}C
Quantity of coal consumed	: 1.8 TPH
Feed water temperature	: 85 ^{0}C
GCV of coal	: 3200 kcal/kg
Enthalpy of saturated steam at 10 kg/cm^2 pressure	: 665 kcal/kg
Enthalpy of feed water	: 85 kcal/kg

Solution :

$$\text{Boiler Efficiency } (\eta) = \frac{8 \times (665 - 85)}{1.8 \times 3200} \times 100$$

$$= 80\% \text{ (on GCV basis)}$$

$$Evaporation\ ratio = \frac{Quantity\ of\ steam\ generated, kg / hr}{Quantity\ of\ fuel\ consumed, kg / hr}$$

$$Evaporation\ ratio = \frac{8000}{1800} = 4.44$$

It should be noted that boiler may not generate 100% saturated dry steam, and there may be some amount of wetness in the steam.

Advantages of direct method:

- Plant people can evaluate the efficiency of boilers quickly
- Requires few parameters for computation
- Needs few instruments for monitoring

The constraints with the direct efficiency method:

- Accuracy of steam/feed water flow measurement and fuel measurement are prone to errors
- Steam quality assessment in case of saturated steam conditions is prone to error as prevalence of wet steam is common in many boilers.
- The reported efficiency gives only values with respect to design but does not indicate any improvement opportunities

The Indirect or Heat Loss Method

Indirect method is also called as heat loss method. The efficiency is arrived at, by subtracting the heat loss fractions from 100. The standards do not include blow down loss in the efficiency determination process. A detailed procedure for calculating boiler efficiency by indirect method is given below along with example.

Boiler reference	:	20 TPH
Steam pressure	:	66 kg/cm^2
Flue gas:		
O_2 in flue gas	=	9 %
CO in flue gas	=	800 ppm
CO_2 in flue gas	=	10.67%
Average flue gas temperature	=	180^{o}C
Atmospheric air:		
Ambient temperature	=	29.3 ^{o}C
Humidity in ambient air	=	0.01977 kg / kg dry air

Fuel analysis:		
Carbon	=	53.65 %
Hydrogen	=	3.25 %
Nitrogen	=	1.11 %
Oxygen	=	8.68 %
Sulphur	=	0.34 %
Moisture	=	14.43 %
Ash content	=	18.54 %
GCV of Coal	=	4291 kcal/kg
Ash analysis:		
Unburnts in bottom ash	=	0.11 %
Unburnts in fly ash	=	4.89 %
GCV of Bottom ash	=	889 kcal/kg
GCV of fly ash	=	395 kcal/kg

Solution :

Boiler efficiency by indirect method

Step – 1 Find theoretical air requirement

Theoretical air required for complete combustion

= $[(11.6\,x\,C)+\{34.8\,x\,(H_2-O_2/8)\}+(4.35\,x\,S)]/100$ kg/kg of coal

= [(11.6 x 53.65) + {34.8 x (3.25 – 8.68/8)} + (4.35 x 0.34)] / 100

= **7.0 kg / kg of coal**

Step – 2 To find Excess air supplied

Actual O_2 measured in flue gas = 9 %

% Excess air supplied (EA) = $\frac{O_2\%}{21-O_2\%}\times 100$

= $\frac{9}{21-9}\times 100$

= **75%**

Step – 3 To find actual mass of air supplied

Actual mass of air supplied = {1 + EA/100} x theoretical air

= {1+ 75/100} x 7

AAS = **12.3 kg/kg of coal**

Step – 4 To find actual mass of dry flue gas

Mass of dry flue gas = Mass of CO_2 in flue gas +Mass of N_2 content in the fuel+ Mass of N_2 in the combustion air supplied + Mass of oxygen in flue gas + Mass of SO_2 in flue gas

$$\text{Mass of dry flue gas} = \frac{0.5365\,x\,44}{12} + 0.0111 + \frac{12.3\,x\,77}{100} + \frac{(12.3-7)\,x\,23}{100} + \frac{0.0034\times 64}{32}$$

= **12.7 kg / kg of coal**

Step – 5 To find all losses

1. % Heat loss in dry flue gas (L_1) = $\dfrac{m\,x\,C_p\,x\,(T_f - T_a)}{GCV\ of\ fuel}\ x\,100$

$$= \frac{12.7\,x\,0.24\,x\,(180-29.3)}{4291}\ x\,100$$

L_1 = **10.7 %**

2. % Heat loss due to formation of water from H_2 in fuel (L_2) = $\dfrac{9\ x\ H_2\ x\ \{584 + C_p\ (T_f - T_a)\}}{GCV\ of\ fuel}\ x\ 100$

$$= \frac{9\ x\ 0.0325x\ \{584 + 0.45\ (180 - 29.3)\}}{4291}\ x\ 100$$

L_2 = **4.4 %**

3. % Heat loss due to moisture in fuel (L_3) = $\dfrac{M\ x\ \{584 + C_p\ (T_f - T_a)\}}{GCV\ of\ fuel}\ x\ 100$

$$= \frac{0.1443\ x\ \{584 + 0.45\ (180 - 29.3)\}}{4291}\ x\ 100$$

L_3 = **2.2 %**

4. % Heat loss due to moisture in air (L_4) = $\dfrac{AAS\ x\ humidity\ x\ C_p\ x\ (T_f - T_a)}{GCV\ of\ fuel}\ x\,100$

$$= \frac{12.3\,x\,0.01977\,x\,0.45\,x\,(180-29.3)}{4291}\ x\,100$$

L_4 = **0.38 %**

5. % Heat loss due to partial conversion of C to CO (L_5) = $\dfrac{\%CO\ x\ C}{\%CO + \%CO_2}\ x\ \dfrac{5654}{GCV\ of\ fuel}\ x\,100$

$$= \frac{0.08\ x\ 0.5365}{0.08 + 11.5}\ x\ \frac{5654}{4291}\ x\,100$$

L_5 = **0.53%**

6. Heat loss due to radiation and convection and other unaccounted losses (L_6)	=	Assumed as **1.0 %**

The actual radiation and convection losses are difficult to assess because of particular emissivity of various surfaces, its inclination, air flow pattern etc. In a relatively small boiler, with a capacity of 10 MW, the radiation and unaccounted losses could amount to between 1% and 2% of the gross calorific value of the fuel; while in a 500 MW boiler, values between 0.2% to 1% are typical. The loss may be assumed appropriately depending on the surface condition.

7. % Heat loss due to unburnt in fly ash

% Ash in coal	=	18.54 %
Unburnts in fly ash	=	4.89 %
GCV of fly ash	=	395 kCal/kg
Amount of fly ash in 1 kg of coal	=	0.0489 x 0.1854
	=	0.0091 kg
Heat loss due to unburnts in fly ash	=	0.0091 x 395
	=	3.59 kCal / kg of coal
% Heat loss due to unburnts in fly ash	=	3.59 x 100 / 4291
L_7	=	**0.084 %**

8. % Heat loss due to unburnt in bottom ash

GCV of bottom ash	=	889 kCal/kg
Unburnts in bottom ash	=	0.11%
Amount of bottom ash in 1 kg of coal	=	0.0011 x 0.1854
	=	0.000204 kg
Heat loss due to unburnts in bottom ash	=	0.000204 x 889
	=	0.18 kCal/kg of coal
% Heat loss due to unburnts in bottom ash	=	0.18 x 100 / 4291
L_8	=	**0.0042 %**

Boiler efficiency by indirect method	=	$100 – (L_1+ L_2+ L_3+ L_4+ L_5+ L_6+ L_7+ L_8)$
	=	100-(10.7 + 4.4 + 2.2 + 0.38 + 0.49 + 1.0 + 0.084 + 0.0042)
	=	100 - 19.3
	=	**80.7 %**

Summary of Heat Balance for Coal Fired Boiler

Input/Output Parameter		kcal / kg of coal	% loss
Heat Input		4291	100
Losses in boiler			
1. Dry flue gas,	L_1	459.1	10.7
2. Loss due to hydrogen in fuel,	L_2	188.8	4.4
3. Loss due to moisture in fuel,	L_3	94.4	2.2
4. Loss due to moisture in air,	L_4	16.3	0.38
5. Partial combustion of C to CO,	L_5	22.7	0.53
6. Surface heat losses,	L_6	42.9	1.0
7. Loss due to Unburnt in fly ash,	L_7	3.6	0.084
8. Loss due to Unburnt in bottom ash,	L_8	0.18	0.0042
Boiler Efficiency = $100 - (L_1 + L_2 + L_3 + L_4 + L_5 + L_6 + L_7 + L_8)$ = **80.7 %**			

2.5 Boiler Water Treatment

Producing quality steam on demand depends on properly managed water treatment to control steam purity, deposits and corrosion. A boiler is the sump of the boiler system. It ultimately receives all of the pre-boiler contaminants. Boiler performance, efficiency, and service life are direct products of selecting and controlling feed water used in the boiler.

When feed water enters the boiler, the elevated temperatures and pressures cause the components of water to behave differently. Most of the components in the feed water are soluble. However, under heat and pressure most of the soluble components come out of solution as particulate solids, sometimes in crystallized forms and other times as amorphous particles. When solubility of a specific component in water is exceeded, scale or deposits develop. The boiler water must be sufficiently free of deposit forming solids to allow rapid and efficient heat transfer and it must not be corrosive to the boiler metal.

Deposit Control

Deposits in boilers may result from hardness contamination of feed water and corrosion products from the condensate and feed water system. Hardness contamination of the feed water may arise due to deficient softener system.

Deposits and corrosion result in efficiency losses and may result in boiler tube failures and inability to produce steam. Deposits act as insulators and slow heat transfer. Large amounts of deposits throughout the boiler could reduce the heat transfer enough to reduce the boiler efficiency significantly. Different type of deposits affects the boiler efficiency differently. Thus it may be useful to analyse the deposits for its characteristics. The insulating effect of deposits causes the boiler metal temperature to rise and may lead to tube-failure by overheating.

Impurities causing deposits

The most important chemicals contained in water that influences the formation of deposits in the boilers are the salts of calcium and magnesium, which are known as hardness salts.

Calcium and magnesium bicarbonate dissolve in water to form an alkaline solution and these salts are known as alkaline hardness. They decompose upon heating, releasing carbon dioxide and forming a soft sludge, which settles out. These are called temporary hardness-hardness that can be removed by boiling.

Calcium and magnesium sulphates, chlorides and nitrates, etc. when dissolved in water are chemically neutral and are known as non-alkaline hardness. These are called permanent hardness and form hard scales on boiler surfaces, which are difficult to remove. Non-alkalinity hardness chemicals fall out the solution due to reduction in solubility as the temperature rises, by concentration due to evaporation which takes place within the boiler, or by chemical change to a less soluble compound.

Silica

The presence of silica in boiler water can rise to formation of hard silicate scales. It can also associate with calcium and magnesium salts, forming calcium and magnesium silicates of very low thermal conductivity. Silica can give rise to deposits on steam turbine blades, after been carried over either in droplets of water in steam, or in volatile form in steam at higher pressures.

Two major types of boiler water treatment are: Internal water treatment and External water treatment.

Internal Water Treatment

Internal treatment is carried out by adding chemicals to boiler to prevent the formation of scale by converting the scale-forming compounds to free-flowing sludges, which can be removed by blow down. This method is limited to boilers, where feed water is low in hardness salts, to low pressures-high TDS content in boiler water is tolerated, and when only small quantity of water is required to be treated. If these conditions are not applied, then high rates of blow down are required to dispose off the sludge. They become uneconomical from heat and water loss consideration.

Different waters require different chemicals. Sodium carbonate, sodium aluminate, sodium phosphate, sodium sulphite and compounds of vegetable or inorganic origin are all used for this purpose. Proprietary chemicals are available to suit various water conditions. The specialist must be consulted to determine the most suitable chemicals to use in each case. Internal treatment alone is not recommended.

External Water Treatment

External treatment is used to remove suspended solids, dissolved solids (particularly the calcium and magnesium ions which are major causes of scale formation) and dissolved gases (oxygen and carbon dioxide).

The external treatment processes available are: ion exchange; demineralization; reverse osmosis and de-aeration. Before any of these are used, it is necessary to remove suspended solids and colour from the raw water, because these may foul the resins used in the subsequent treatment sections.

Methods of pre-treatment include simple sedimentation in settling tanks or settling in clarifiers with aid of coagulants and flocculants. Pressure sand filters, with spray aeration to remove carbon dioxide and iron, may be used to remove metal salts from bore well water.

The first stage of treatment is to remove hardness salt and possibly non-hardness salts. Removal of only hardness salts is called softening, while total removal of salts from solution is called demineralization.

The processes are:

Ion-exchange process (Softener Plant)

In ion-exchange process, the hardness is removed as the water passes through bed of natural zeolite or synthetic resin and without the formation of any precipitate. The simplest type is 'base exchange' in which calcium and magnesium ions are exchanged for sodium ions. After saturation regeneration is done with sodium chloride. The sodium salts being soluble, do not form scales in boilers. Since base exchanger only replaces the calcium and magnesium with sodium, it does not reduce the TDS content, and blowdown quantity. It also does not reduce the alkalinity.

Softening reaction
$Na_2R + Ca(HCO_3)_2 \leftrightarrow CaR + 2\ Na(HCO_3)$

Regeneration reaction
$CaR + 2\ NaCl \leftrightarrow Na_2R + CaCl_2$

Demineralization is the complete removal of all salts. This is achieved by using a "cation" resin, which exchanges the cations in the raw water with hydrogen ions, producing hydrochloric, sulphuric and carbonic acid. Carbonic acid is removed in degassing tower in which air is blown through the acid water. Following this, the water passes through an "anion" resin which exchanges anions with the mineral acid (e.g. sulphuric acid) and forms water. Regeneration of cations and anions is necessary at intervals using, typically, mineral acid and caustic soda respectively. The complete removal of silica can be achieved by correct choice of anion resin.

Ion exchange processes can be used for almost total demineralization if required, as is the case in large electric power plant boilers

De-aeration

In de-aeration (Figure 2.9), dissolved gases, such as oxygen and carbon dioxide, are expelled by preheating the feed water before it enters the boiler.

All natural waters contain dissolved gases in solution. Certain gases, such as carbon dioxide and oxygen, greatly increase corrosion. When heated in boiler systems, carbon dioxide (CO_2) and oxygen (O_2) are released as gases and combine with water (H_2O) to form carbonic acid, (H_2CO_3).

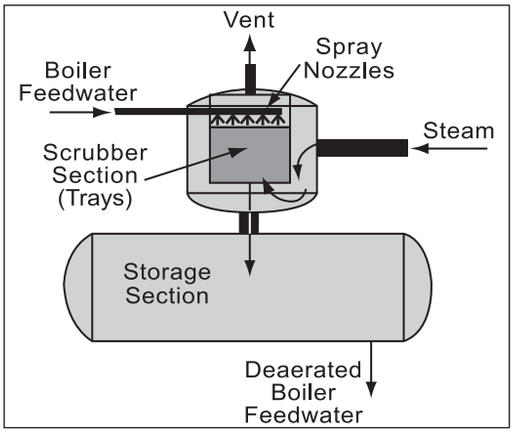


Figure 2.9 Deaerator

Removal of oxygen, carbon dioxide and other non-condensable gases from boiler feed water is vital to boiler equipment longevity as well as safety of operation. Carbonic acid corrodes metal reducing the life of equipment and piping. It also dissolves iron (Fe) which when returned to the boiler precipitates and causes scaling on the boiler and tubes. This scale not only contributes to reducing the life of the equipment but also increases the amount of energy needed to achieve heat transfer.

De-aeration can be done by mechanical de-aeration, by chemical de-aeration or by both together.

Mechanical de-aeration

Mechanical de-aeration for the removal of these dissolved gases is typically utilized prior to the addition of chemical oxygen scavengers. Mechanical de-aeration is based on Charles' and Henry's laws of physics. Simplified, these laws state that removal of oxygen and carbon dioxide can be accomplished by heating the boiler feed water, which reduces the concentration of oxygen and carbon dioxide in the atmosphere surrounding the feed water. Mechanical de-aeration can be the most economical. They operate at the boiling point of water at the pressure in the de-aerator. They can be of vacuum or pressure type.

The vacuum type of de-aerator operates below atmospheric pressure, at about 82^{o}C, can reduce the oxygen content in water to less than 0.02 mg/litre. Vacuum pumps or steam ejectors are required to maintain the vacuum.

The pressure-type de-aerators operates by allowing steam into the feed water through a pressure control valve to maintain the desired operating pressure, and hence temperature at a minimum of 105^{o}C. The steam raises the water temperature causing the release of O_2 and CO_2 gases that are then vented from the system. This type can reduce the oxygen content to 0.005 mg/litre.

Where excess low-pressure steam is available, the operating pressure can be selected to make use of this steam and hence improve fuel economy. In boiler systems, steam is preferred for de-aeration because:

- Steam is essentially free from O_2 and CO_2
- Steam is readily available
- Steam adds the heat required to complete the reaction.

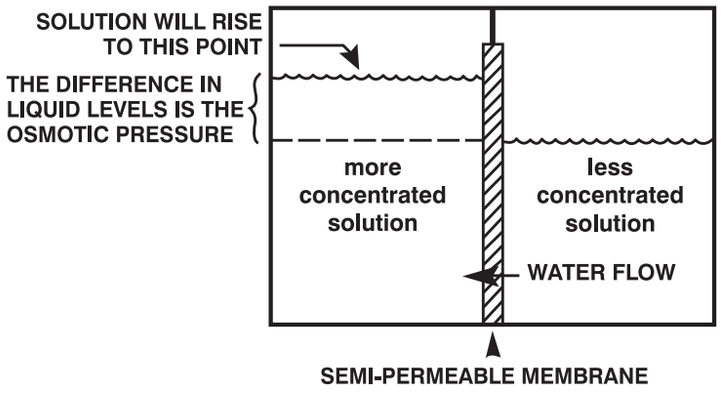
Chemical de-aeration

While the most efficient mechanical deaerators reduce oxygen to very low levels (0.005 mg/litre), even trace amounts of oxygen may cause corrosion damage to a system. Consequently, good operating practice requires removal of that trace oxygen with a chemical oxygen scavenger such as sodium sulfite or hydrazine. Sodium sulphite reacts with oxygen to form sodium sulphate, which increases the TDS in the boiler water and hence increases the blow down requirements and make-up water quality. Hydrazine reacts with oxygen to form nitrogen and water. It is invariably used in high pressures boilers when low boiler water solids are necessary, as it does not increase the TDS of the boiler water.

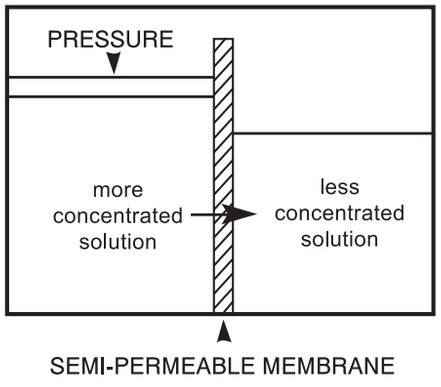
Reverse Osmosis

When solutions of differing concentrations are separated by a semi-permeable membrane, water from less concentrated solution passes through the membrane to dilute the liquid of high concentration, which is called osmosis. If the solution of high concentration is pressurized, the process is reversed and the water from the solution of high concentration flows to the weaker solution. This is known as reverse osmosis.

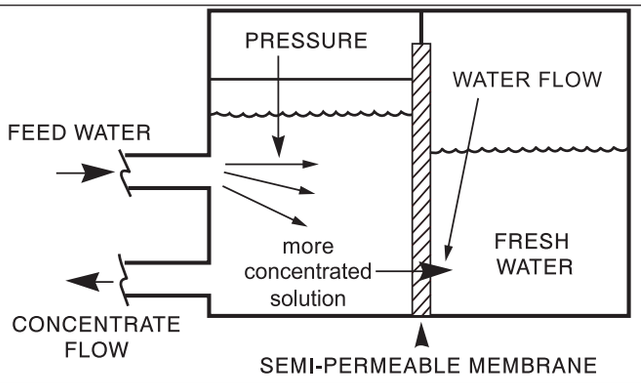
The quality of water produced depends upon the concentration of the solution on the high-pressure side and pressure differential across the membrane. This process is suitable for waters with very high TDS, such as sea water.



The semi permeable nature of the membrane allows the water to pass much more readily than the dissolved minerals. Since the water in the less concentrated solution seeks to dilute the more concentrated solution, the water passage through the membrane generates a noticeable head difference between the two solutions. This head difference is a measure of the concentration difference of the two solutions and is referred to as the osmotic pressure difference.



When a pressure is applied to the concentrated solution which is great that the osmotic pressure difference, the direction of water passage through the membrane *is reversed* and the process that we refer to as reverse osmosis is established. That is, the membrane's ability to selectively pass water is unchanged, only the direction of the water flow is changed.



The feed water and *concentrate (reject stream) ports* illustrates a continuously operating RO system.

Figure 2.10 Reverse Osmosis

Recommended boiler and feed water quality

The impurities found in boiler water depend on the untreated feed water quality, the treatment process used and the boiler operating procedures. As a general rule, the higher the boiler operating pressure, the greater will be the sensitivity to impurities. Recommended feed water and boiler water limits are shown in Table 2.2 and Table 2.3.

Table 2.2 Recommended Feed Water Limits

Factor	Upto 20 kg/cm^2	21 – 39 kg/cm^2	40- 59 kg/cm^2
Total iron (max) ppm	0.05	0.02	0.01
Total copper (max) ppm	0.01	0.01	0.01
Total silica (max) ppm	1.0	0.3	0.1
Oxygen (max) ppm	0.02	0.02	0.01
Hydrazine residual ppm	-	-	-0.02-0.04
pH at 25 ^{0}C	8.8-9.2	8.8-9.2	8.2-9.2
Hardness, ppm	1.0	0.5	-

Table 2.3 Recommended Boiler Water Limits (IS 10392, Year 1982)

Factor	Upto 20 kg/cm^2	21 - 39 kg/cm^2	40 - 59 kg/cm^2
TDS, ppm	3000-3500	1500-2500	500-1500
Total iron dissolved solids ppm	500	200	150
Specific electrical conductivity at 25^{0}C (mho)	1000	400	300
Phosphate residual ppm	20-40	20-40	15-25
pH at 25^{0}C	10-10.5	10-10.5	9.8-10.2
Silica (max) ppm	25	15	10

2.6 Boiler Blow Down

When water is boiled and steam is generated, any dissolved solids contained in the water remain in the boiler. If more solids are put in with the feed water, they will concentrate and may eventually reach a level where their solubility in the water is exceeded and they deposit from the solution. Above a certain level of concentration, these solids encourage foaming and cause carryover of water into the steam. The deposits also lead to scale formation inside the boiler, resulting in localized overheating and finally causing boiler tube failure.

It is, therefore, necessary to control the level of concentration of the solids and this is achieved by the process of 'blowing down', where a certain volume of water is blown off and is automatically replaced by feed water - thus maintaining the optimum level of total dissolved solids (TDS) in the boiler water. Blow down is necessary to protect the surfaces of the heat exchanger in the boiler. However, blow down can be a significant source of heat loss, if improperly carried out. The maximum amount of total dissolved solids (TDS) concentration permissible in various types of boilers is given in Table 2.4.

Table 2.4 Recommended TDS Levels for Various Industrial Process Boilers

	Boiler Type	Maximum TDS (ppm)*
1.	Lancashire	10,000 ppm
2.	Smoke and water tube boilers (12 kg/cm^2)	5,000 ppm
3.	Low pressure Water tube boiler	2000-3000
3.	High Pressure Water tube boiler with superheater etc.	3,000 - 3,500 ppm
4.	Package and economic boilers	3,000 ppm
5.	Coil boilers and steam generators	2000 (in the feed water

Note: Refer guidelines specified by manufacturer for more details
*parts per million

Conductivity as Indicator of Boiler Water Quality

Since it is tedious and time consuming to measure total dissolved solids (TDS) in boiler water system, conductivity measurement is used for monitoring the overall TDS present in the boiler. A rise in conductivity indicates a rise in the "contamination" of the boiler water.

Conventional methods for blowing down the boiler depend on two kinds of blow down - intermittent and continuous

Intermittent Blow Down

The intermittent blown down is given by manually operating a valve fitted to discharge pipe at the lowest point of boiler shell to reduce parameters (TDS or conductivity, pH, Silica and Phosphates concentration) within prescribed limits so that steam quality is not likely to be affected. In intermittent blow down, a large diameter line is opened for a short period of time, the time being based on a thumb rule such as "once in a shift for 2 minutes".

Intermittent blow down requires l*arge* short-term increases in the amount of feed water put into the boiler, and hence may necessitate larger feed water pumps than if continuous blow down is used. Also, TDS level will be varying, thereby causing fluctuations of the water level in the boiler due to changes

in steam bubble size and distribution which accompany changes in concentration of solids. Also substantial amount of heat energy is lost with intermittent blow down.

Continuous Blow Down

There is a steady and constant dispatch of small stream of concentrated boiler water, and replacement by steady and constant inflow of feed water. This ensures constant TDS and steam purity at given steam load. Once blow down valve is set for a given conditions, there is no need for regular operator intervention.

Even though large quantities of heat are wasted, opportunity exists for recovering this heat by blowing into a flash tank and generating flash steam. This flash steam can be used for pre-heating boiler feed water or for any other purpose (see Figure 2.11 for blow down heat recovery system). This type of blow down is common in high-pressure boilers.

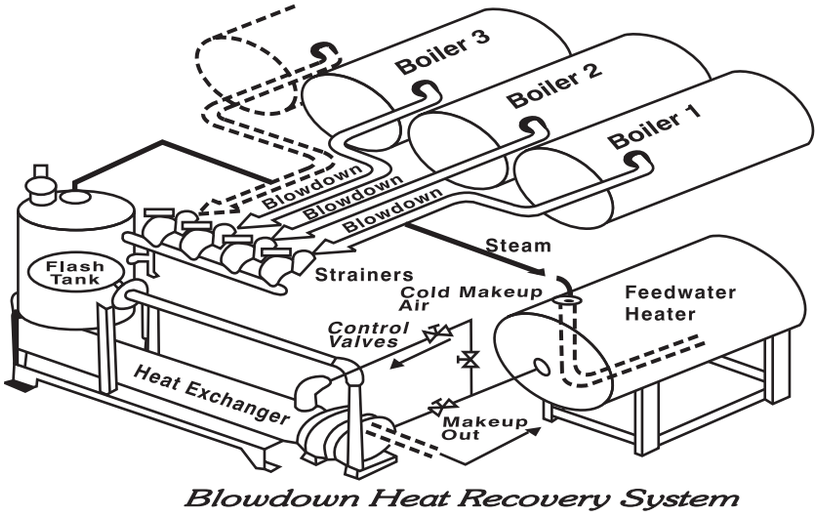


Figure 2.11 Blowdown Heat Recovery System

Blow down calculations

The quantity of blow down required to control boiler water solids concentration is calculated by using the following formula:

$$\text{Blow down (\%)} = \frac{[\text{Feed water TDS x \% Make up water}]\times 100}{(\text{Maximum Permissible TDS in Boiler water - Feed water TDS})}$$

If maximum permissible limit of TDS as in a package boiler is 3000 ppm, percentage make up water is 10% and TDS in feed water is 300 ppm, then the percentage blow down is given as:

$$= \frac{300\times(10/100)}{3000-300}\times 100 = 1.11\ \%$$

If boiler evaporation rate is 3000 kg/hr then required blow down rate is:

$$= \frac{3000\times 1.1}{100} = 33.3\ \text{kg/hr}$$

Benefits of Blow Down

Good boiler blow down control can significantly reduce treatment and operational costs that include:

- Lower pre-treatment costs
- Less make-up water consumption
- Reduced maintenance downtime
- Increased boiler life
- Lower consumption of treatment chemicals

2.7 Improving Boiler Availability

i) Reducing Boiler Tube Leakage

Tube failures in boilers lead to reduction in availability and call for continuous efforts to minimize the tube leakages.

Tube failures can be attributed to design, manufacturing, operational, maintenance and ageing related aspects. The tube failures mechanisms can be classified as Mechanical, chemical, metallurgical & in adequate quality compliance.

Mechanical causes

Mechanical damage of tubes takes place due to fly ash erosion, steam impingement from soot blowers, falling clinker and fuel particles. The Mechanical damage increases the stress level leading to tube failure.

Chemical causes

Water side and fire side corrosion failure of tubes occurs due to phenomenon like caustic gouging, hydrogen embrittlement, pitting, and stress corrosion cracking while fire side corrosion is often due to high as well as low temperature effects

Metallurgical causes

In high temp components like superheaters, reheaters creep damage occurs due to overheating wherein the tube material loses its strength and failure occurs by stress rupture. Weld joint failures by cracking and fatigue failures by vibration, thermal and corrosion phenomenon are other reasons for metallurgical related tube failures.

Quality assurance gaps

Tube failures are also known to occur due to inadequate quality assurance procedures during design/ manufacturing of boiler tubes. Due care during material procurement, adoption of healthy fabrication practices, good operation and maintenance practices in tube manufacturing would help to minimize tube leakages and achieve high availability.

ii) Soot Blowing and Soot Deposit Reduction

One of the important by-products of the combustion of solid fuels is ash. During combustion, some of the mineral constituents and compounds can be in a molten or plastic state. The ash constituents can then deposit on the heat-absorbing surfaces causing slagging and fouling. If ash reaches the heat-absorbing surfaces at a temperature near its softening temperature, the resulting deposits are likely to be porous and can be removed by soot blowing. Slagging is the deposition of molten, partially fused deposits on the furnace walls and the radiant superheaters exposed to radiant heat. Fouling is the deposition of more loosely bonded deposits on the heat-absorbing surfaces in the convective path. Soot blowers using steam are generally used for removing ash deposits on fouled tube surfaces, while water lances or water canons using thermal shock by room-temperature water may be needed to clean the slagged tube surfaces.

A Soot blower is a system for removing the soot that is deposited on the furnace tubes of a boiler during combustion. Various types of soot blowers such as Wall Blowers, Long Retractable Blowers and Air Heater Blowers are used for the cleaning. Steam is normally used as a medium for blowing away the soot.

The soot which sticks to the heating surfaces insulates them from the heat, reducing their efficiency. Soot must be blown off to maximize heat transfer in the boiler. Elevated stack temperatures may indicate excessive soot build-up. High exit gas temperatures at normal excess air indicate poor heat transfer performance.

Stack temperature should be checked and recorded regularly as an indicator of soot deposits. When the flue gas temperature raises by about 20 ^{0}C above the normal temperature for a newly cleaned boiler, soot deposits are the main concern. It is estimated that 3 mm of soot can cause an increase in fuel consumption by 2.5% due to increased flue gas temperatures.

iii) Preservation of Boiler

The internal surfaces of the boiler are prone to corrosion by leftover water after operation or by atmospheric oxygen when they are out of service. Hence boiler preservation is required to protect the internal surfaces from corrosion. There are two methods of preservation a) wet method and b) dry method.

Boilers which are kept ready for standby service and may be required for sudden demands of operations are preserved by wet method. Dry method is recommended for boilers which are scheduled to be kept out of service for a long period and are not expected to be put into operation at short notice.

a) Wet method

In the wet method the boiler is filled to the normal level with water at a pH of 10.5 to 11. Hydrazine to the extent of 200 ppm is to be dosed with the water. The unit is to be steamed in service to ensure uniform concentration of boiler water throughout the unit and to eliminate dissolved oxygen from water. Sodium sulphite (Na_2SO_3), which acts as a de-oxygenerator, can also be used as an alternative to hydrazine and the sulphite concentration has to be maintained at 300-400 ppm.

Analysis of boiler water should be carried out frequently. If the hydrazine concentration in water happens to drop below 50 ppm, the water in the drum should be lowered to the normal operating level and an appropriate quantity of chemicals should be dosed to bring back 200 the concentration of hydrazine or sodium sulphite. The boiler should be steamed to circulate chemicals to uniform concentration.

b) Dry method

This is mainly done to preserve the internal surfaces of the boiler. The boiler is filled with deaerated water and fired to raise the pressure to 7 kg/cm^2. The firing is now stopped and the pressure is allowed to drop to 2-3 kg/cm^2, the water is then drained out of the boiler. All the drain pockets and manhole doors are kept open. Hot air is blown through the blowers to completely dry the internal surfaces. After thorough drying trays containing silica gel or activated alumina (0.25 kg/m^2 of heating surface) should be placed inside the drum. Alternatively, anhydrous calcium chloride or un-slaked lime (0.098 kg/m^2 of heating surface) may be used as a desiccant. The pressure parts are then closed air tight to avoid any moisture/air leaking in. the inspection should be carried out once in a month and the desiccant trays replenished accordingly.

Another method of dry preservation is to keep the boiler under positive pressure by using pure nitrogen.

iv) Cold Start up and Shutdown Procedures:

a) Cold start up procedure

Starting up of boiler should not be speeded up, and it normally takes about four hours to complete. Strict adherence to the manufacturer's recommended procedures will make the boiler operation more efficient and safer. A typical cold start procedure is given below.

- Check that the boiler is properly closed up (especially after repairs).
- Check physically that all appropriate valves are shut or open for safe starting of the boiler.
- Boiler is to be filled to slightly below normal level.
- Boiler treatment chemicals may now be added to the boiler water.
- Check and clear the furnace of any flammable materials.
- Ensure that the boiler uptakes passage is clear.
- Pre-purging of furnace for a specified amount of time is necessary to clear the gas-side of flammable gases to avoid a staring explosion.
- On a cold boiler, firing-up must not be speeded up too much in order not to overstrain the boiler material unnecessarily by quick, uneven temperature raises.

Keep the boiler vents open until a heavy steam jet is flowing out (until a boiler pressure of about 1 bar is reached).

b) Boiler shut down procedures

During normal operations, the boiler water chemistry is carefully controlled so that the dissolved/ suspended material is conditioned to prevent hard deposits on boiler metal. These dissolved/suspended

solids are maintained in suspension by water circulation and the action of the treatment chemicals. When a boiler is shut down or drained, this material (sludge) may settle and bake on tube surfaces; it may become so adherent that mechanical (turbining) chemical cleaning may be required. A typical shutdown procedure to avoid such problem, is given below.

Three to five days before a scheduled shut down, increase the blow down by 50%.

- If possible, increase the alkalinity to at least 500 ppm without causing foaming or carryover.
- Due to the increased blow down rate, the feed rate of the scale inhibitor and oxygen scavenger must be increased so as to maintain the normal boiler water residuals.
- Increase the sludge conditioner level in the boiler water by 50 to 100%.

During the last twenty-four hours before shut down, decrease the continuous blow down and increase the manual blow down.

- Frequent short bottom blows are better than fewer longer blows.
- Generally it is sufficient to hold each mud drum blow down valve open for about 5-10 seconds every one to two hours.
- Once the load is dropped from the boiler, include the header blow downs as part of the manual blow down procedure.
- When the load is dropped from the boiler, continue bottom blow downs until boiler is cool and safe to work on.
- As soon as possible after the boiler is opened, wash down the boiler watersides, preferably with soft water.

2.8 Thermic Fluid Heaters

Water/steam is used as heat carrier in many heating applications. However, at high temperatures, steam requires a corresponding high operating pressure. Industrial heating systems, a high temperature level is often a great advantage, and establishing this with steam can be very cumbersome and expensive in some cases.

In thermic fluid heaters, a special type of oil-synthetic / mineral -is used as heat carrier. This fluid can be heated up to 300^{o}C at atmospheric pressure. In comparison steam would require a pressure of 85 bars to obtain this temperature.

There are several advantages in using thermic fluids compared to steam systems. The most obvious advantages are as follows.

- High temperature operation at atmospheric pressure
- Optional temperature level set points
- No supply of treatment of heat water and hence no heat loss due to condensate flash steam
- No risk of corrosion
- Easy to operate

The heaters are made with coils of seamless tubes. The thermal fluid is heated during the flow through

the tubes (Refer Figure 2.12). The heat is transferred to the fluid as radiant heat in the combustion chamber, where the inner cylindrical tube coil and a flat tube coil form the chamber wall and the bottom respectively. The hot thermic fluid is circulated to various process equipments such as dryers, heaters, deodouriser etc. where it gives up the heat. The return oil at a temperature 10 to 20 ^{0}C less comes back to the thermic fluid heater to get heated up again. The circulation is carried out by a thermic fluid circulation pump.

The thermic fluid heater operates between two temperature ranges. Once the upper limit is reached the burner is switched OFF or goes into the low fire mode. In the case of solid fuel fired system the ID fan switches OFF on reaching the upper limit. When the temperature reaches the lower limit due to heat transfer in the process, the burners come ON again and in case of solid fuels, the ID fan comes ON again.

Since the thermic fluid heaters operate at a high temperature (250 – 300 ^{0}C), the leaving exhaust gas temperatures are more than the fluid temperature. Hence, the heat loss through the flue gas is a major component of fuel losses. This offers potential for heat recovery if there is a suitable application. The capacity of the thermic fluid heater is specified in terms of Lakh kilo Calories per hour or Million kilo Calories per hour.

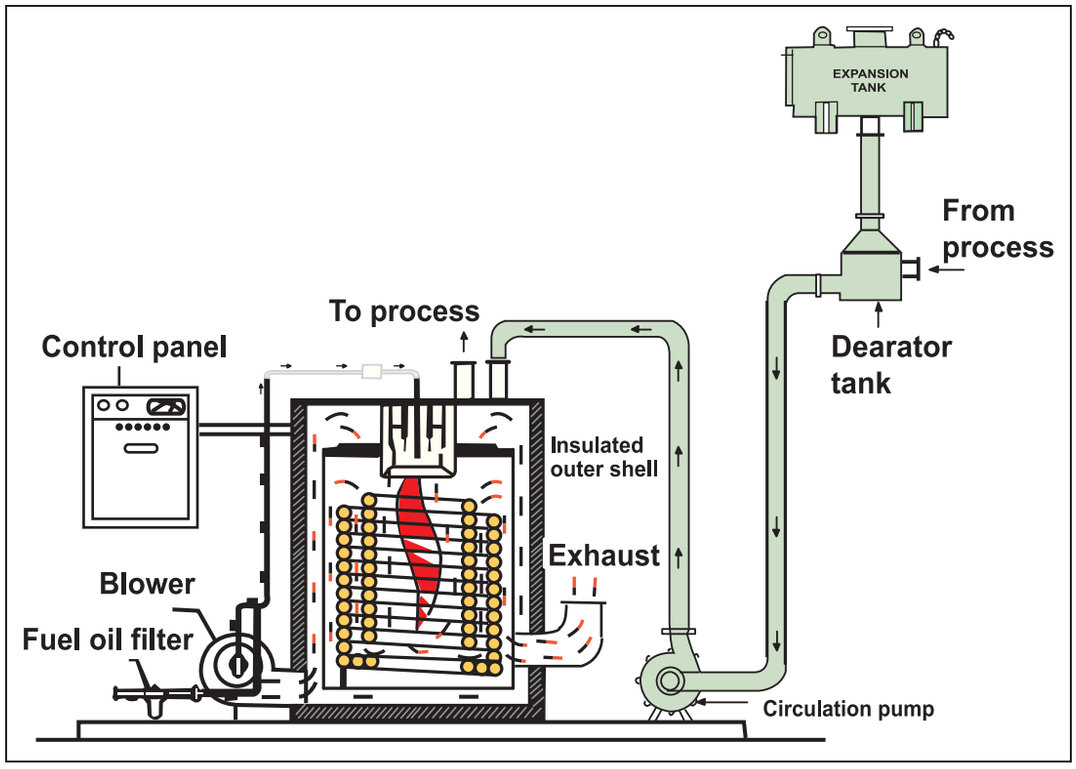


Figure 2.12 Thermic Fluid Heater

2.9 Energy Conservation Opportunities

The various energy efficiency opportunities in boiler system can be related to combustion, heat transfer, avoidable losses, high auxiliary power consumption, water quality and blow down. Examining the following factors can indicate if a boiler is being run to maximize its efficiency:

1. Stack Temperature

The stack temperature should be as low as possible. However, it should not be so low that water vapor in the exhaust condenses on the stack walls. This is important in fuels containing significant sulphur as low temperature can lead to sulphur dew point corrosion. Stack temperatures greater than 200 °C indicates potential for recovery of waste heat. It also indicate the scaling of heat transfer/recovery equipment and hence the urgency of taking an early shut down for water / flue side cleaning.

2. Feed Water Preheating using Economiser

Typically, the flue gases leaving a modern 3-pass shell boiler are at temperatures of 200 to 300 ^{0}C. Thus, there is a potential to recover heat from these gases. The flue gas exit temperature from a boiler is usually maintained at a minimum of 200 ^{0}C, so that the sulphur oxides in the flue gas do not condense and cause corrosion in heat transfer surfaces. When a clean fuel such as natural gas, LPG or gas oil is used, the economy of heat recovery must be worked out, as the flue gas temperature may be well below 200 ^{0}C .

The potential for energy saving depends on the type of boiler installed and the fuel used. For a typically older model shell boiler, with a flue gas exit temperature of 260 ^{0}C, an economizer could be used to reduce it to 200 ^{0}C, increasing the feed water temperature by 15 ^{0}C. Increase in overall thermal efficiency would be in the order of 3%. For a modern 3-pass shell boiler firing natural gas with a flue gas exit temperature of 140 ^{0}C a condensing economizer would reduce the exit temperature to 65 ^{0}C increasing thermal efficiency by 5%.

3. Combustion Air Preheat

Combustion air preheating is an alternative to feedwater heating. In order to improve thermal efficiency by 1%, the combustion air temperature must be raised by 20 ^{0}C. Most gas and oil burners used in a boiler plant are not designed for high air preheat temperatures.

Modern burners can withstand much higher combustion air preheat, so it is possible to consider such units as heat exchangers in the exit flue as an alternative to an economizer, when either space or a high feed water return temperature make it viable.

4. Incomplete Combustion

Incomplete combustion can arise from a shortage of air or surplus of fuel or poor distribution of fuel. It is usually obvious from the colour or smoke, and must be corrected immediately.

In the case of oil and gas fired systems, CO or smoke (for oil fired systems only) with normal or high excess air indicates burner system problems. A more frequent cause of incomplete combustion is the poor mixing of fuel and air at the burner. Poor oil fires can result from improper viscosity, worn tips, carbonization on tips and deterioration of diffusers or spinner plates.

With coal firing, unburned carbon can comprise a big loss. It occurs as grit carry-over or carbon-in-ash and may amount to more than 2% of the heat supplied to the boiler. Non uniform fuel size could be one of the reasons for incomplete combustion. In chain grate stokers, large lumps will not burn out completely, while small pieces and fines may block the air passage, thus causing poor air distribution.

In sprinkler stokers, stoker grate condition, fuel distributors, wind box air regulation and over-fire systems can affect carbon loss. Increase in the fines in pulverized coal also increases carbon loss.

5. Excess Air Control

The Table 2.5 gives the theoretical amount of air required for combustion of various types of fuel. Excess air is required in all practical cases to ensure complete combustion, to allow for the normal variations in combustion and to ensure satisfactory stack conditions for some fuels. The optimum excess air level for maximum boiler efficiency occurs when the sum of the losses due to incomplete combustion and loss due to heat in flue gases is minimum. This level varies with furnace design, type of burner, fuel and process variables. It can be determined by conducting tests with different air fuel ratios.

Table 2.5 Typical Combustion Data – Common Boiler Fuels for Illustrative Purpose

Fuel	kg of air req./kg of fuel	kg of flue gas/kg of fuel	m^3 of flue/ kg of fuel	Theoretical CO_2% in dry flue gas	CO_2% in flue gas achieved in practice
Solid Fuels					
Bagasse	3.2	3.43	2.61	20.65	10-12
Coal (bituminous)	10.8	11.7	9.40	18.70	10-13
Lignite	8.4	9.10	6.97	19.40	9 -13
Paddy Husk	4.6	5.63	4.58	19.8	14-15
Wood	5.8	6.4	4.79	20.3	11.13
Liquid Fuels					
Furnace Oil	13.90	14.30	11.50	15.0	9-14
LSHS	14.04	14.63	10.79	15.5	9-14

Typical values of excess air supplied for various fuels are given in Table – 2.6

Table 2.6 Excess Air Levels for Different Fuels

Fuel	Type of Furnace or Burners	Excess Air (% by wt)
Pulverised coal	Completely water-cooled furnace for slag-tap or dry-ash removal	15-20
	Partially water-cooled furnace for dry-ash removal	15-40
Coal	Spreader stoker	30-60
	Water-cooler vibrating-grate stokers	30-60
	Chain-grate and traveling-grate stokers	15-50
	Underfeed stoker	20-50
Fuel oil	Oil burners, register type	15-20
	Multi-fuel burners and flat-flame	20-30
Natural gas	High pressure burner	5-7
Wood	Dutch over (10-23% through grates) and Hofft type	20-25
Bagasse	All furnaces	25-35
Black liquor	Recovery furnaces for draft and soda-pulping processes	30-40

Controlling excess air to an optimum level always results in reduction in flue gas losses; for every 1% reduction in excess air there is approximately 0.6% rise in efficiency.

Various methods are available to control the excess air:

- Portable oxygen analysers and draft gauges can be used to make periodic readings to guide the operator to manually adjust the flow of air for optimum operation. Excess air reduction up to 20% is feasible.
- The most common method is the continuous oxygen analyzer with a local readout mounted draft gauge, by which the operator can adjust air flow. A further reduction of 10-15% can be achieved over the previous system.
- The same continuous oxygen analyzer can have a remote controlled pneumatic damper positioner, by which the readouts are available in a control room. This enables an operator to remotely control a number of firing systems simultaneously.

The most sophisticated system is the automatic stack damper control, whose cost is really justified only for large systems.

6. Radiation and Convection Heat Loss

The external surfaces of a shell boiler are hotter than the surroundings. The surfaces thus lose heat to the surroundings depending on the surface area and the difference in temperature between the surface and the surroundings.

The heat loss from the boiler shell is normally a fixed energy loss, irrespective of the boiler output. With modern boiler designs, this may represent only 1.5% on the gross calorific value at full rating, but will increase to around 6%, if the boiler operates at only 25 percent output.

Repairing or augmenting insulation can reduce heat loss through boiler walls and piping.

7. Automatic Blowdown Control

Uncontrolled continuous blowdown is very wasteful. Automatic blowdown controls can be installed that sense and respond to boiler water conductivity and pH. A 10% blow down in a 15 kg/cm^2 boiler results in 3% efficiency loss.

8. Reduction of Boiler Steam Pressure

This is an effective means of reducing fuel consumption, if permissible, by as much as 1 to 2%. Lower steam pressure gives a lower saturated steam temperature and without stack heat recovery, a similar reduction in the temperature of the flue gas temperature results.

Steam is generated at pressures normally dictated by the highest pressure / temperature requirements for a particular process. In some cases, the process does not operate all the time, and there are periods when the boiler pressure could be reduced. The energy manager should consider pressure reduction

carefully, before recommending it. Adverse effects, such as an increase in water carryover from the boiler owing to pressure reduction, may negate any potential saving. Pressure should be reduced in stages, and no more than a 20 percent reduction should be considered.

9. Variable Speed Control for Fans, Blowers and Pumps

Variable speed control is an important means of achieving energy savings. Generally, combustion air control is effected by throttling dampers fitted at forced and induced draft fans. Though dampers are simple means of control, they lack accuracy, giving poor control characteristics at the top and bottom of the operating range. In general, if the load characteristic of the boiler is variable, the possibility of replacing the dampers by a VSD should be evaluated.

10. Effect of Boiler Loading on Efficiency

The maximum efficiency of the boiler does not occur at full load, but at about two-thirds of the full load. If the load on the boiler decreases further, efficiency also tends to decrease. At zero output, the efficiency of the boiler is zero, and any fuel fired is used only to supply the losses. The factors affecting boiler efficiency are:

- As the load falls, so does the value of the mass flow rate of the flue gases through the tubes. This reduction in flow rate for the same heat transfer area, reduced the exit flue gas temperatures by a small extent, reducing the sensible heat loss.
- Below half load, most combustion appliances need more excess air to burn the fuel completely. This increases the sensible heat loss.

In general, efficiency of the boiler reduces significantly below 25% of the rated load and as far as possible; operation of boilers below this level should be avoided.

11. Proper Boiler Scheduling

Since, the optimum efficiency of boilers occurs at 65-85% of full load, it is usually more efficient, on the whole, to operate a fewer number of boilers at higher loads, than to operate a large number at low loads.

12. Boiler Replacement

The potential savings from replacing a boiler depend on the anticipated change in overall efficiency. A change in a boiler can be financially attractive if the existing boiler is :

- Old and inefficient
- Not capable of firing cheaper substitution fuel
- Over or under-sized for present requirements
- Not designed for ideal loading conditions

The feasibility study should examine all implications of long-term fuel availability and company growth plans. All financial and engineering factors should be considered. Since boiler plants traditionally have a useful life of well over 25 years, replacement must be carefully studied.

Solved Example:

An oil fired boiler is generating 30 T/hr Steam and operates for 8000 hrs/year. The TDS in boiler feed water was reduced from 500 ppm to 200 ppm. The maximum permissible limit of TDS in the boiler is 3000 ppm and make up water is 10%.

Temperature of the blow down water is 170°C and boiler feed water temperature is 40°C. GCV of fuel is 10000 kcal/kg and efficiency of the boiler is 80%.

Calculate the savings in fuel oil per annum due to reduction in the blow down.

Ans:

Blow down % = Feed water TDS * % make up water * 100 / (maximum permissible TDS in boiler water – Feed water TDS)

Initial blow down = 500 * 10 / (3000 – 500)
Initial blow down = 2.0 %

Improved blow down = 200 * 10 / (3000 – 200)
Improved blow down = 0.71 %

Reduction in blow down = 2.0 – 0.71
Reduction in blow down = 1.29 %

Reduction in blow down = 1.29 * 30 * 1000 / 100
Reduction in blow down = 387 kg/hr

Heat savings = 387 * 1 * (170 – 40)
Heat savings = 50310 kcal/hr
Fuel Oil saving = 50310 / (10000 * 0.8) = 6.28 kg/hr
= 6.28 * 8000 hr = **50.24 MT / annum**

QUESTIONS	
Objective Type Questions	
1.	A system in which a package type boiler operates with a FD fan is called a) Natural draft b) Forced draft c) Induced draft d) Balanced draft
2.	Steam generation in a boiler is 26 tonnes in 2 hours. Fuel consumption in the same period is 2 tonnes. The evaporation ratio is a) 10 b) 12 c) 13 d) 26
3.	Economizer in a boiler is used to extract heat from flue gases for pre-heating a) blow down b) fuel oil c) air for combustion of fuel d) feed water
4.	Concentration of solids in boiler drum is controlled by a) steam venting b) blow down c) steam trap d) excess air
5.	Major heat loss in an oil fired boiler is accounted by a) surface radiation loss b) stack loss c) un-burnt carbon loss d) blow down loss
6.	A system in which the boiler operates with a FD and ID fan is called a) Natural draft b) Forced draft c) Induced draft d) Balanced draft
7.	Suitable atomizing viscosity of fuel oil for use in LAP/MAP burners is a) 300 Redwood Seconds-1 b) 100 Redwood Seconds-1 c) 400 Redwood Seconds-1 d) 600 Redwood Seconds-1
8.	The presence of silica in the boiler water causes a) steam carryover b) corrosion c) scale formation d) none of the above
9.	Removal of dissolved gases from the boiler feed water is called a) degasification b) deaeration c) deoxidation d) none of the above
10.	In flue gas the theoretical CO_2 is 15.5% and measured CO_2 is 11% by volume. The percentage of excess air will be a) 40.9% b) 38.7 % c) 240.9 % d) 140.9 %
Short Type Questions	
S-1	A boiler operator is trying to reduce excess air level from 30% to 10% in a coal fired boiler. Describe what will most likely happen to the CO, CO_2 concentration as well as amount of unburnt carbon in the flue gas. Explain your answer.
S-2	The evaporation ratio of a coal fired boiler is 4. Estimate boiler efficiency if steam enthalpy is 640 kcal/kg; feed water temperature is 55^0C and Calorific Value of coal is 4000 kcal/kg.
S-3	In a boiler drum the permissible TDS is 2000 ppm, permissible make up water is 10 per cent of feed water and TDS of feed water is 200 ppm. Find out the blow down percentage?
S-4	The efficiency of a boiler on GCV basis is 82%. The fuel contains 0.5 % moisture and 11 % hydrogen. The GCV of fuel is 10,300 kcal/kg. What is the boiler efficiency on the basis of net calorific value?

S-5	For a oil containing 9% hydrogen (GCV = 10,200 kcal/kg), estimate the percentage of sensible and latent heat loss due to evaporation of water formed due to hydrogen in the fuel, if the flue gas temperature is 180ºC and combustion air temperature is 40ºC. (latent heat of vapours = 584 kcal/kg, specific heat of vapours = 0.45 kcal/kg/^{0}C)
	Long Type Questions
L-1	Evaluate the option of boiler replacement for the following boiler with a new boiler of 84% efficiency. The cost of new boiler is Rs 30.00 lakh. Data of present boiler: • Average steam generation from the boiler: 5000 kg/h • Fuel used: furnace oil • Enthalpy gained by the steam in boiler: 600 kcal/kg of steam • Cost of furnace oil: Rs 15000 per ton (Rs. 15 per kg) • Gross calorific value of the fuel: 10000 kcal/kg • Annual operating hours of the boiler: 6000 h • Boiler efficiency: 80%
L-2	A chemical plant has an AFBC boiler with the following specifications: Boiler capacity : 70 TPH Boiler pressure : 60 kg/cm^2 Steam temperature : 500^{o}C Fuel fired : coal with 35% ash content GCV of coal : 4100 kcal/kg Theoretical air for combustion : 5.6 kg/kg of coal Hydrogen in fuel : 4% Specific heat of flue gas : 0.24 kcal/kg^{o}C Specific heat of superheated water vapor in the flue gas : 0.45 kcal/kg^{o}C Operating parameters are given below Flue gas exhaust temperature : 160^{o}C Excess air : 30% Feed water temperature : 105^{o}C Radiation and other losses : 8% Ambient temperature : 30^{o}C (i) Calculate the Boiler Efficiency using indirect method on GCV basis. (ii) If the feed water temperature is 110^{o}C and the steam is produced at 60 kg/ cm^2 & 500^{o}C, what is the hourly coal consumption? Total heat of steam at 60 kg/ cm^2 & 500^{o}C is 817 kcal/kg.

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Chapter 3

Steam System

Learning Objectives

In this chapter you will learn about

- Heat transfer modes in boiler
- Properties of steam
- Steam distribution system
- Types of steam traps and its characteristics
- Efficient utilization of steam
- Condensate and flash steam recovery
- Energy efficiency opportunities in steam systems

3. STEAM SYSTEM

Properties of Steam, Steam Distribution, PRDS, Steam Pipe Insulation, Thermo Compressor, Dryers, Proper Selection, Operation and Maintenance of Steam Traps, Condensate Recovery and Pumping System, Flash Steam Recovery, Efficient Steam Utilization and Energy Saving Opportunities.

3.1 Introduction

Steam has been a popular mode of conveying energy since the industrial revolution. Steam is used for generating power and also used in process industries such as sugar, paper, fertilizer, refineries, petrochemicals, chemical, food, synthetic fiber and textiles. The following characteristics of steam make it so popular and useful to the industry:

- Highest specific heat and latent heat
- Highest heat transfer coefficient
- Easy to control and distribute
- Cheap and inert

3.2 Properties of Steam

Water can exist in the form of solid, liquid and gas as ice, water and steam respectively. If heat energy is added to water, its temperature rises until a value is reached at which the water can no longer exist as a liquid. We call this the "saturation" point and with any further addition of energy, some of the water will boil off as steam. This evaporation requires relatively large amounts of energy, and while it is being added, the water and the steam released are both at the same temperature. Equally, if steam is made to release the energy that was added to evaporate it, then the steam will condense and water at same temperature will be formed.

Liquid Enthalpy

Liquid enthalpy is the "Enthalpy" (heat energy) in the water when it has been raised to its boiling point to produce steam, and is measured in kcal/kg, its symbol is h_f. (also known as "Sensible Heat")

The heat required to change the temperature of a substance is called its **sensible heat**.

If 1 kg of water in a vessel at 25 °C i.e. containing heat value of 25 kcal is heated by adding 75 kcal, the water is brought to boiling point of 100 °C.

Enthalpy of Evaporation (Heat Content of Steam)

The Enthalpy of evaporation is the heat energy to be added to the water (when it

To change the water to steam, an additional 540 kcal would be required. This quantity of heat required to change a chemical from the liquid to the gaseous state is called **latent heat**.

has been raised to its boiling point) in order to change it into steam. There is no change in temperature, the steam produced is at the same temperature as the water from which it is produced, but the heat energy added to the water changes its state from water into steam at the same temperature.

When the steam condenses back into water, it gives up its enthalpy of evaporation, which it had acquired on changing from water to steam. The enthalpy of evaporation is measured in kcal/kg. Its symbol is h_{fg}. Enthalpy of evaporation is also known as latent heat.

For a boiler is operating at a pressure of 8 kg/cm^2, steam saturation temperature is 170^{o}C and steam enthalpy or total heat of dry saturated steam is given by:
$h_f + h_{fg} = 171.35 + 489.46 = 660.81$ kcal/kg.

If the same steam contains 4% moisture, the total heat of steam is given by:
171.35 + 0.96 x 489.46 = 641.23 kcal/kg.

The temperature at which water boils, also called as boiling point or **saturation temperature** increases as the pressure increases. When water under pressure is heated, its saturation temperature rises above 100 ^{0}C. From this, it is evident that as the steam pressure increases, the usable heat energy in the steam (enthalpy of evaporation), which is given up when the steam condenses, actually decreases. The total heat of dry saturated steam or enthalpy of saturated steam is given by sum of the two enthalpies $h_f + h_{fg}$ (Refer Table 3.1 and Figure 3.1). When the steam contains moisture the total heat of steam will be $h_g = h_f + X\ h_{fg}$ where X is the dryness fraction.

The temperature of saturated steam is the same as the water from which it is generated, and corresponds to a fixed and known pressure. Superheat is the addition of heat to dry saturated steam without increase in pressure. The temperature of superheated steam, expressed as degrees above saturation corresponding to the pressure, is referred to as the degrees of **superheat**.

The Steam Phase Diagram

The data provided in the steam tables can also be expressed in a graphical form. Figure 3.1 illustrates the relationship between the enthalpy and the temperature at various different pressures, and is known as a phase diagram.

As water is heated from 0 ^{o}C to its saturation temperature, its condition follows the saturated liquid line until it has received all of its liquid enthalpy, h_f, (**A - B**).

If further heat continues to be added, it then changes phase to saturated steam and continues to increase in enthalpy while remaining at saturation temperature, h_{fg}, (**B - C**).

As the steam/water mixture increases in dryness, its condition moves from the saturated liquid line to the saturated vapour line. Therefore at a point exactly halfway between these two states, the dryness fraction (X) is 0.5. Similarly, on the saturated vapour line the steam is 100% dry.

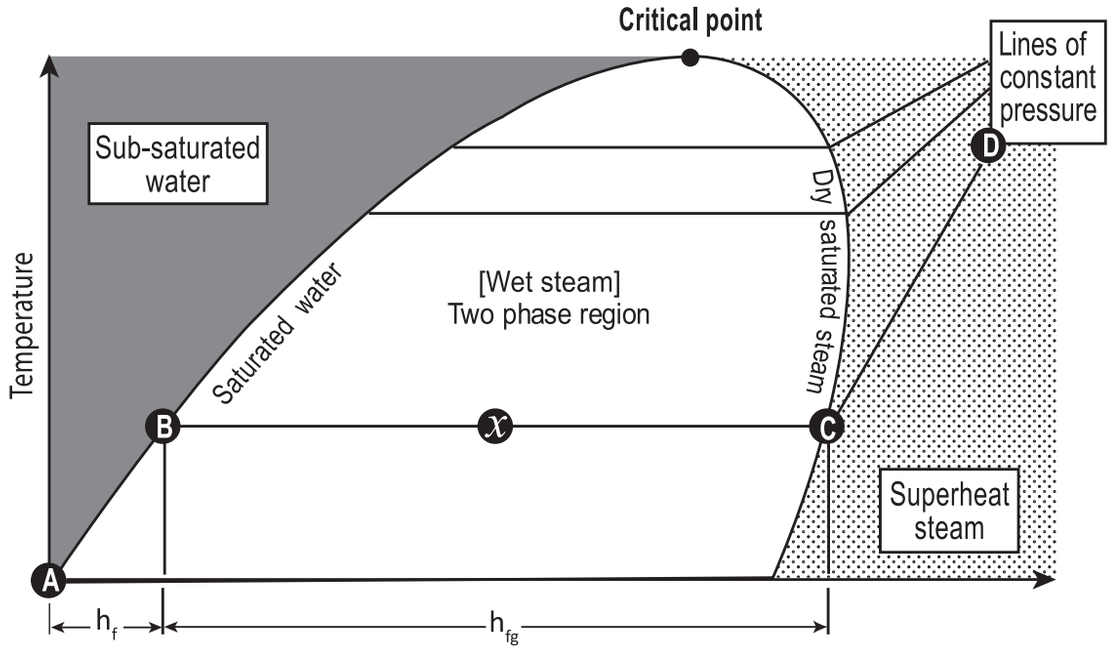


Figure 3.1 Steam Phase Diagram

Once it has received all of its enthalpy of evaporation, it reaches the saturated vapour line. If it continues to be heated after this point, the temperature of the steam will begin to rise as superheat is imparted (**C - D**).

The saturated liquid and saturated vapour lines enclose a region in which a steam/water mixture exists - wet steam. In the region to the left of the saturated liquid line only water exists, and in the region to the right of the saturated vapour line only superheated steam exists.

The point at which the saturated liquid and saturated vapour lines meet is known as the critical point. As the pressure increases towards the critical point the enthalpy of evaporation decreases, until it becomes zero at the critical point. This suggests that water changes directly into saturated steam at the critical point.

Above the critical point only gas may exist. The gaseous state is the most diffuse state in which the molecules have an almost unrestricted motion, and the volume increases without limit as the pressure is reduced.

The critical point is the highest temperature at which liquid can exist. Any compression at constant temperature above the critical point will not produce a phase change. Compression at constant temperature below the critical point however, will result in liquefaction of the vapour as it passes from the superheated region into the wet steam region. The critical point occurs at 374.15°C and 221.2 bar (a) for steam. Above this pressure the steam is termed supercritical and no well-defined boiling point applies.

Table 3.1 Extract From the Steam Tables					
Pressure (kg/cm^2)	Temperature ^oC	Enthalpy in kcal/kg			Specific volume (m^3/kg)
		Water (h_f)	Evaporation (h_{fg})	Steam (h_g)	
1	100	100.09	539.06	639.15	1.673
2	120	119.92	526.26	646.18	0.901
3	133	133.42	517.15	650.57	0.616
4	143	143.70	509.96	653.66	0.470
5	151	152.13	503.90	656.03	0.381
6	158	159.33	498.59	657.92	0.321
7	164	165.67	493.82	659.49	0.277
8	170	171.35	489.46	660.81	0.244

3.3 Steam Distribution

The steam distribution system is the essential link between the steam generator and the steam user. Whatever the source, an efficient steam distribution system is essential if steam of the right quality and pressure is to be supplied, in the right quantity, to the steam using equipment. Installation and maintenance of the steam system are important issues, and must be considered at the design stage.

As steam condenses in a process, flow is induced in the supply pipe. Condensate has a very small volume compared to the steam, and this causes a pressure drop, which causes the steam to flow through the pipes. The steam generated in the boiler must be conveyed through pipe work to the point where its heat energy is required. Initially there will be one or more main pipes, or 'steam mains', which carry steam from the boiler in the general direction of the steam using plant. Smaller branch pipes can then carry the steam to the individual pieces of equipment. A typical steam distribution system is shown in Figure 3.2.

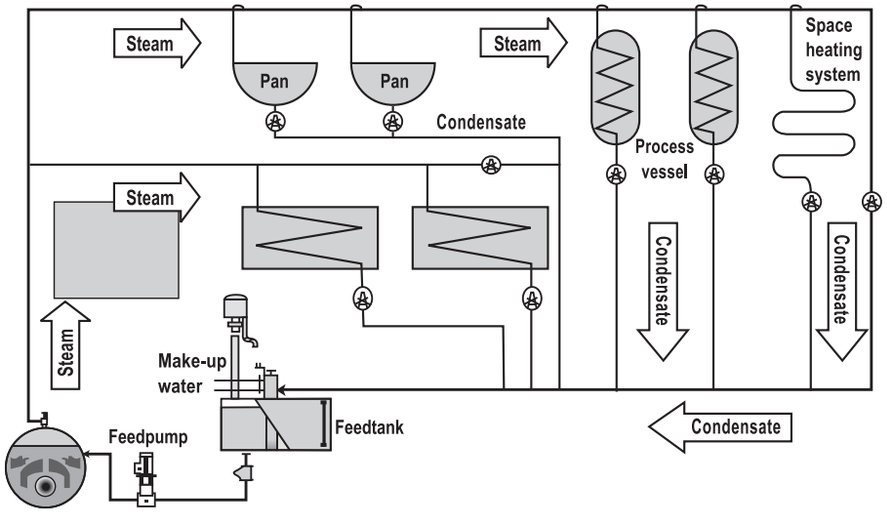


Figure 3.2 Steam Distribution System

The working pressure

The distribution pressure of steam is influenced by a number of factors, but is limited by:

- The maximum safe working pressure of the boiler
- The minimum pressure required at the plant

As steam passes through the distribution pipe work, it will inevitably lose pressure due to:

- Frictional resistance within the pipe work
- Condensation within the pipe work as heat is transferred to the environment.

Therefore allowance should be made for this pressure loss when deciding upon the initial distribution pressure. Having determined the working pressure, the following factors have to be considered in the design of a good Steam Distribution System.

- General Layout
- Pipe sizing and Design
- Pressure Reducing and Des-superheating Station
- Air Venting
- Steam Pipe Insulation

General Layout

General layout and location of steam consuming equipment is of great importance in efficient distribution of steam. Steam pipes should be laid by the shortest possible distance rather than to follow a building layout or road etc. However, this may come in the way of aesthetic design and architect's plans and a compromise may be necessary while laying new pipes.

Steam Pipe Sizing and Design

Any modification and alteration in the existing steam piping, for supplying higher quality steam at right pressure and quantity must consider the following points:

Pipe Sizing

The objective of the steam distribution system is to supply steam at the correct pressure to the point of use. It follows, therefore, that pressure drop through the distribution system is an important feature. Proper sizing of steam pipelines help in minimizing pressure drop. The velocities for various types of steam are:

Superheated	50-70 m/sec
Saturated	30-40 m/sec
Wet or Exhaust	20-30 m/sec

For fluid flow to occur, there must be more energy at Point 1 than Point 2 (see Figure 3.3). The difference in energy is used to overcome frictional resistance between the pipe and the flowing fluid.

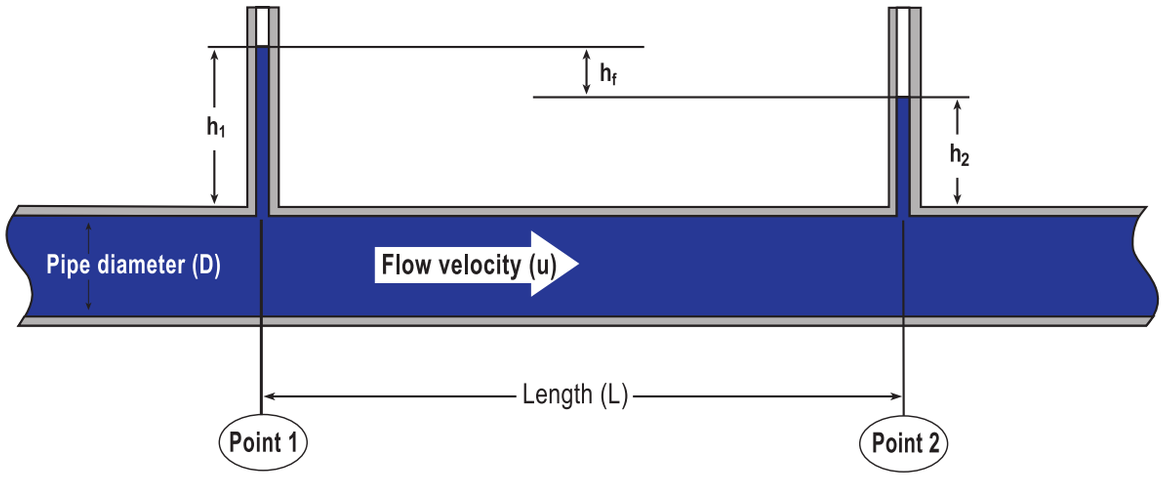


Figure 3.3 Pressure drop in steam pipes

Source: www.spiraxsarco.com

This is illustrated by the equation

$$h_f = \frac{4 f L u^2}{2 g D}$$

Where,
h_f = Head loss to friction (m)
f = Friction factor (dimensionless)
L = Length (m)
u = Flow velocity (m/s)
g = Gravitational constant (9.81 m/s²)
D = Pipe diameter (m)

It is useful to remember that:

- Head loss to friction (h_f) is proportional to the velocity squared (u^2).
- The friction factor (f) is an experimental coefficient which is affected by factors including:
  - The Reynolds Number (which is affected by velocity).
  - The reciprocal of velocity².

Because the values for 'f' are quite complex, they are usually obtained from charts.

Example – Pressure drop calculation

Determine the difference in pressure between two points 1 km apart in a 150 mm bore horizontal pipework system. The water flowrate is 45 m³/h at 15°C and the friction factor for this pipe is taken as 0.005.

$$Velocity\,(m/s) = \frac{Volume\ flowrate\,(m^3/s)}{Cross\,\text{sec}tional\ area\,(m^2)}$$

$$Velocity\,(m/s) = \frac{45\,(m^3/hr)\times 4}{3600\,s/hr\times\Pi\times 0.15^{\,2}}$$

$$Velocity = 0.71\,m/s$$

$$h_f = \frac{4\,f\,L\,u^2}{2\,g\,D}$$

$$h_f = \frac{4\times 0.005\times 1000\times 0.71^2}{2\times 9.81\times 0.15}$$

$$h_f = 3.43\,m \approx 0.34\,bar$$

Example – Determining the pipe size

A process requires 5 000 kg/h of dry saturated steam at 7 kg/cm² (g). For the flow velocity not to exceed 25 m/s, determine the pipe size.

$$Flow\,velocity = 25\,m/s$$

$$Specific\ volume\ at\ 7\,kg/cm^2 = 0.24\,m^3/kg$$

$$Mass\ flow\,rate = 5000\,kg/hr\ or\ 1.389\,kg/s$$

$$Volumetric\ flowrate = 1.389\times 0.24$$

$$= 0.333\,m^3/s$$

Therefore, using:

$$Cross\,\text{sec}tional\ area = \frac{Volumetric\ flowrate}{Flow\,velocity}$$

$$\frac{\Pi\times D^2}{4} = \frac{Volumetric\ flowrate}{Flow\,velocity}$$

$$D^2 = \frac{4 \times Volumetric\ flowrate}{\prod \times Flow\ velocity}$$

$$D = \sqrt{\frac{4 \times Volumetric\ flowrate}{\prod \times Flow\ velocity}}$$

$$D = \sqrt{\frac{4 \times 0.333}{\prod \times 25}}$$

$$D = 0.130\,m\ or\ 130\,mm$$

Since the steam velocity must not exceed 25 m/s, the pipe size must be at least 130 mm; the nearest commercially available size, 150 mm, would be selected.

In practice whether for water pipes or steam pipes, a balance is drawn between pipe size and pressure loss. The steam piping should be sized, based on permissible velocity and the available pressure drop in the line. Selecting a higher pipe size will reduce the pressure drop and thus the energy cost. However, higher pipe size will increase the initial installation cost. By use of smaller pipe size, even though the installation cost can be reduced, the energy cost will increase due to higher-pressure drop. It is to be noted that the pressure drop change will be inversely proportional to the 5th power of diameter change. Hence, care should be taken in selecting the optimum pipe size.

Guide for proper drainage and layout of steam lines:

1. The steam mains should be run with a falling slope of not less that 125 mm for every 30 metres length in the direction of the steam flow.
2. Drain points should be provided at intervals of 30-45 metres along the main.
3. Drain points should also be provided at low points in the mains and where the steam main rises. Ideal locations are the bottom of expansion joints and before reduction and stop valves.
4. Drain points in the main lines should be through an equal tee connection only.
5. It is preferable to choose open bucket or TD traps on account of their resilience.
6. The branch lines from the mains should always be connected at the top. Otherwise, the branch line itself will act as a drain for the condensate.
7. Insecure supports as well as an alteration in level can lead to formation of water pockets in steam, leading to wet steam delivery. Providing proper vertical and support hangers helps overcome such eventualities.
8. Expansion loops are required to accommodate the expansion of steam lines while starting from cold.
9. To ensure dry steam in the process equipment and in branch lines, steam separators can be installed as required.

Pressure Reducing De-superheating Station

A reduction in steam pressure through a pressure reducing valve (PRV) is an isenthalpic process. Saturated steam when reduced to a lower pressure results in superheated steam. Since the process

requires only saturated steam, desuperheating is often required, to compensate for superheat gained in PRV application due to isenthalpic expansion. Pressure reduction and desuperheating of steam or conditioning of steam is done in many process industries to suit process requirement. This is due to the fact that steam is produced in a boiler economically at higher pressure and temperature. Generally, the temperature of the steam after desuperheating will be closer to saturation temperature for heat transfer applications.

The desuperheating of steam is done by spraying water through a spray nozzle into a pipe. Normally, the desuperheating is done by automatic control system, using a control device for spraying water, which takes the feed back from the temperature control loop.

Proper Air Venting

When steam is first admitted to a pipe after a period of shutdown, the pipe is full of air. Further, amounts of air and other non-condensable gases will enter with the steam, although the proportions of these gases are normally very small compared with the steam. When the steam condenses, these gases will accumulate in pipes and heat exchangers. Precautions should be taken to discharge them. The consequence of not removing air is a lengthy warming up period, reduction in plant efficiency and process performance. Air in the steam system will also affect the system temperature. Air will exert its own pressure within the system, which will add to the pressure of the steam to give a total pressure. Therefore, the actual steam pressure and temperature of the steam-air mixture will be lower than that shown by pressure gauge.

Automatic air vents for steam systems (which operate on the same principle as thermostatic steam traps) should be fitted above the condensate level so that only air or steam-air mixtures can reach them. The best location for them is at the end of the steam mains as shown in Figure 3.4.

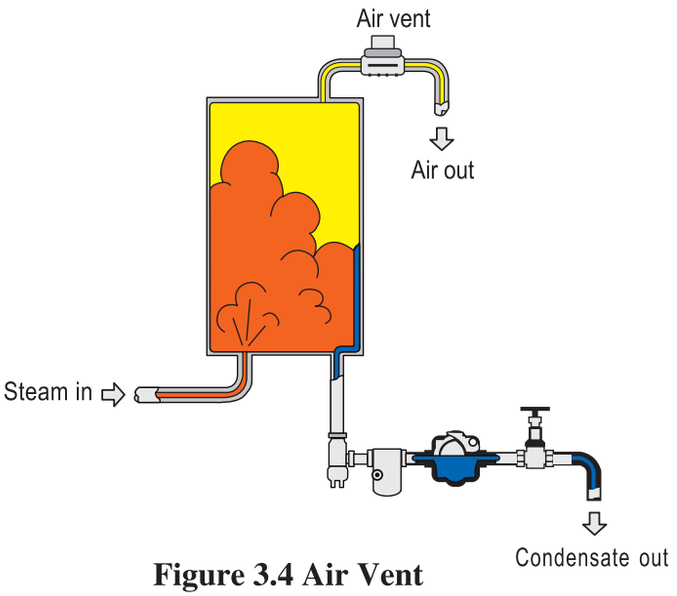


Figure 3.4 Air Vent

In addition to air venting at the end of a main, air vents should also be fitted:

- In parallel with an inverted bucket trap or, in some instances, a thermodynamic trap. These traps are sometimes slow to vent air on start-up.
- In awkward steam spaces (such as at the opposite side to where steam enters a jacketed pan).

- Where there is a large steam space (such as an autoclave), and a steam/air mixture could affect the process quality.

Example – Effect of air in steam system

If 20% of air is entrained in a steam system at 5 kg/cm^2 (g) then the effect of air will be as follows

Steam quality = 80% Steam + 20% Air
Pressure = 0.80 x 5 + 0.20 x 5
= 4 kg/cm^2 (g) + 1 kg/cm^2 (g)
(steam) (air)

Temp. of steam at 5 kg/cm^2 (g) = 158 ^{0}C
Temp. of vapour mixture is = 152 ^{0}C (equivalent to steam at 4 kg/cm^2 (g))

Steam Pipe Insulation

The insulation of steam conveying pipes and the steam consuming equipment is very essential to retard the flow of heat from the system to the environment. Broadly the purpose of steam pipe insulation is as follows

- Conserve energy by reducing heat loss
- Facilitate temperature control of a process
- Prevent condensation of steam
- Prevent or reduce damage to pipe from exposure to fire or corrosive atmospheres
- Control surface temperature for personal protection and comfort

The following table 3.2 indicates the effect of insulating bare pipes

Table 3.2 Effect of Insulation on Steam Pipes

Pipe Size, inch	Economic Insulation Thickness, mm	Radiation Losses* (kW/m)	
		Insulated	Uninsulated
1/2	15	125	692
2	25	243	1820
4	40	298	2942
12	50	588	7614

*** Comparison of Radiation Losses (Pipe Surface Temperature at 150 ^{o}C)**

Heat can be lost due to radiation from steam pipes. As an example while lagging steam pipes, it is common to see leaving flanges uncovered. An uncovered flange is equivalent to leaving 0.6 metre of pipe line unlagged. If a 0.15 m steam pipe diameter has 5 uncovered flanges, there would be a loss of heat equivalent to wasting 5 tons of coal or 3000 litres of oil a year. This is usually done to facilitate

checking the condition of flange but at the cost of considerable heat loss. The remedy is to provide easily detachable insulation covers, which can be easily removed when necessary. The various insulating materials used are cork, Glass wool, Rock wool and Asbestos.

Effect of insulation of flanges: 12 Flanges of 150 mm diameter.
Heat loss in the following 2 cases:

Case (I) – Bare flanges
Case (II) – Flanges with 50 mm insulation and aluminum cladding

Parameter	Unit	Case (I)	Case (II)
Heat Loss	kcal/Year	36,300	4,100
Steam Loss	kg/Year	68	3.2
Fuel Loss	kg/Year	55	0.26
Energy Saving Potential	Rs./Year	60	2.8

Note: Calculation procedure to find out the economic thickness of insulation is given in chapter-5: Insulation and Refractories.

3.4 Efficient Steam Utilization

In a steam system the major scope for improving energy efficiency lies in the utilization part. As shown in Figure 3.5 the steam generation and distribution efficiencies are fairly high at more than 80% whereas the utilization efficiency is only 47%.

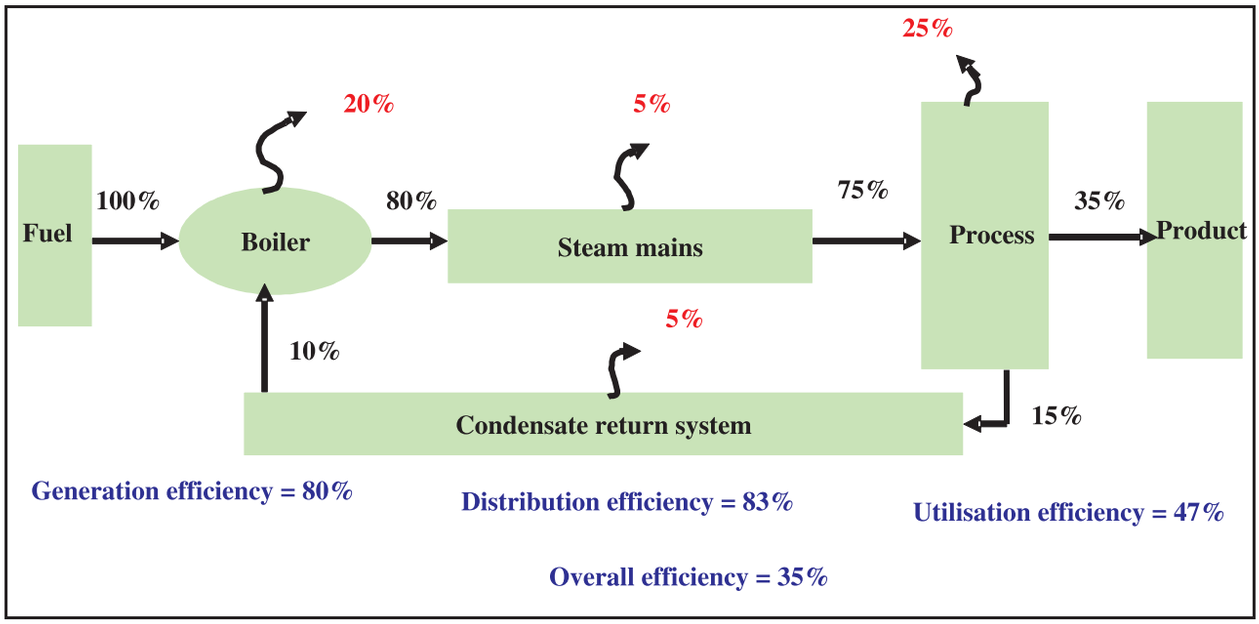


Figure 3.5 Steam System Efficiency

When viewed from the standpoint of being a heat medium, steam has superior properties not offered by other heat mediums. Among those, the following two are the most notable:

- Provides even heating
- Provides rapid heating

In the case of saturated steam, if the steam pressure is known then the steam temperature may be determined. Pressure changes instantaneously within a space. When saturated steam condenses, it condenses at the saturation temperature, and the saturated water (condensate) formed is of the same temperature as the saturated steam. This means that if the pressure at the heat transfer surface (the jacket or coil interior of the equipment) is held at a constant, continuous heating will be able to take place at the same temperature at every part of the heat transfer surface.

The amount of the heat transfer is indicated by the heat transfer coefficient (= film coefficient of heat transfer). The unit is [W/m² K].

W = J/sec, so if heat exchange takes place on the same heat transfer surface area and with the same temperature difference, the larger the heat transfer rate the shorter the time required for heating.

The rough values for the heat transfer rates of hot water and steam are as follows:

- The rate at which heat is transferred to the heat transfer surface of a heat exchanger using hot water as the heat source:
  1000 — 6000 [W/m² K]
- The rate at which heat is transferred to the heat transfer surface of a heat exchanger using steam as the heat source:
  6000 — 15000 [W/m² K]

In actual heating situations, the heat transfer process will be a combination of the mechanism of heat transfer within the walls of the heat exchanger and the mechanism of heat transfer from the wall surface of the heat exchanger to the product being heated (Figure 3.6). Evaluations of heating must use the overall heat transfer coefficient [W/m²K] to indicate this combination. This coefficient varies greatly from heat exchanger to heat exchanger, but even so, steam heating shows numbers 1.5 – 2 times those of hot water heating.

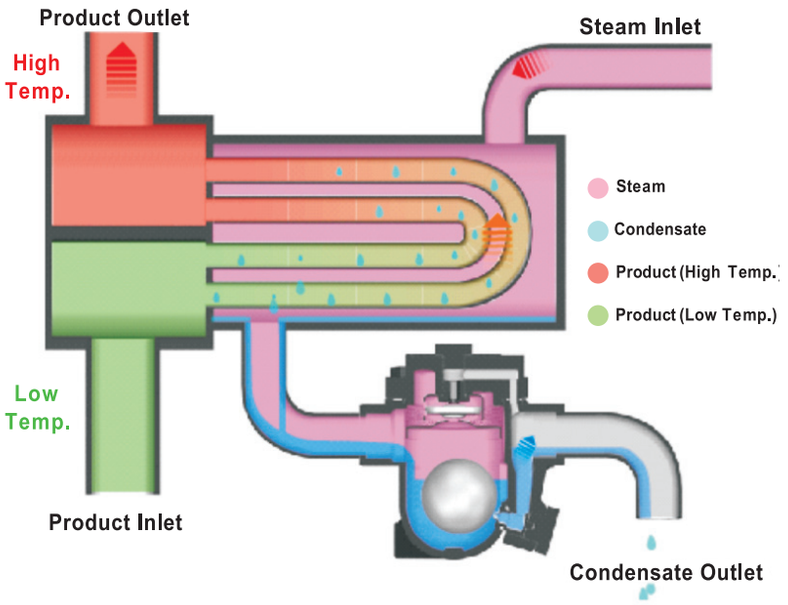


Figure 3.6 Heat Transfer with Steam

Steam provides rapid heating because the transfer of heat caused by the process of condensing. The latent heat contained in steam is released in the instant the steam condenses into liquid phase. The amount of latent heat released is 2 – 5 times greater than the amount of sensible heat in the hot water (saturated water) after condensation. This latent heat is released instantaneously and is transferred by means of a heat exchanger to the product being heated.

In contrast, hot water and oil are used in convective heating, which does not involve a phase change. Instead, the heat medium reduces its own temperature in order to transfer heat to the product being heated. A mainstream in industry is the use of forced-convection by means of equipment such as a pump to create the flow against the heat transfer surface.

Example – Steam Utilization

1) A milk evaporator uses a steam jacketed kettle, in which milk is batch-processed at atmospheric pressure. The kettle has a 680 kg per batch capacity. Milk is heated from a temperature of 26 ^{o}C to 100 ^{o}C, where 25% of its mass is then driven off as vapor. Determine the amount of 1 kg/cm^2 (g) steam required per batch, not including the heating of the kettle itself. (Specific heat of milk is 0.90 kcal/kg ^{o}C). The latent heat of steam at 1 kg/cm^2 is 525 kcal/kg.

Solution:

Quantity of water evaporated from milk = 680 x 0.25
= 170 kg /batch

Heat required to raise the temperature of milk from 26 to 100 ^{o}C
= 680 x 0.9 x (100 – 26)
= 45288 kcal/batch

Amount of heat required to evaporate 170 kg of water
= 170 x 540
= 91800 kcal/batch

Total heat required = 45288 + 91800
= 137088 kcal/batch

Quantity of steam required = 137088/525
= 261 kg/batch

Thermocompressor

In many of the steam utilization equipment where condensate comes out at high pressure, a major portion of it flashes into low pressure steam which goes wasted. Using a thermocompressor (Figure 3.7) it becomes feasible to compress this low pressure steam by high pressure steam and reuse it as a medium pressure steam in the process. The major energy in steam is in its latent heat value and thus thermocompressing would give a large improvement in waste heat recovery.

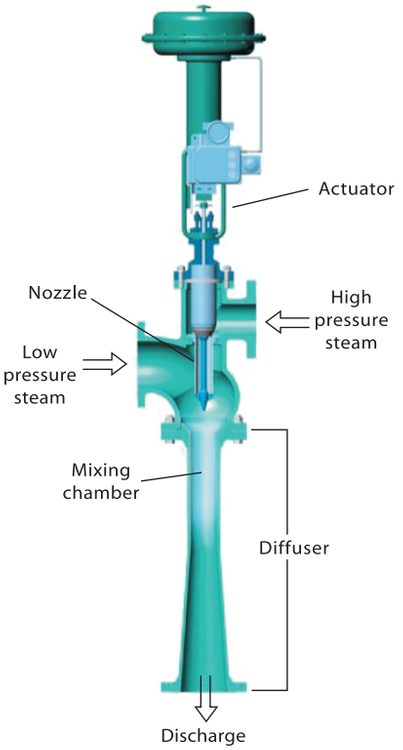


Figure 3.7 Thermocompressor

Thermocompressors are designed to accurately mix lower-pressure steam with higher-pressure steam. The higher-pressure motive steam entrains the lower pressure steam and increases its pressure. The motive steam is introduced through the nozzle of the thermocompressor. As the nozzle opens, the high velocity motive steam draws the lower-pressure steam into the thermocompressor body. An exchange of momentum occurs as the steam flows are mixed and the mixed flow is accelerated to high velocity with a uniform profile in the mixing chamber of the thermocompressor. As the mixed flow enters the diffuser section, the diffuser flow area gradually increases to allow the velocity of the mixed flow to be reduced. As the velocity is reduced, the steam pressure increases. At the end of the diffuser, the discharge steam pressure is higher than the lower-pressure suction flow entering the thermocompressor. A figure of thermocompressor is shown in Figure 3.7. A typical application is in evaporators where the boiling steam is recompressed and used as heating steam.

Advantages of thermocompressors

- No condensation losses take place
- Thermal efficiency of the system is extremely high
- Entrainment of low pressure steam results is substantial savings
- No moving parts and hence maintenance needs are minimal
- No major operational charges
- Low space requirement
- Insensitive to fouling
- High operating reliability

Dryers

Drying is a process by which a liquid (commonly water) is removed from a material. This is usually achieved by applying heat, typically steam and/or the flow of carrier gas (commonly air) through or over the surface of the material (Refer Figure 3.8). The objective of drying is to form a product that meets a water-content specification, so the amount of water removed depends on the desired product.

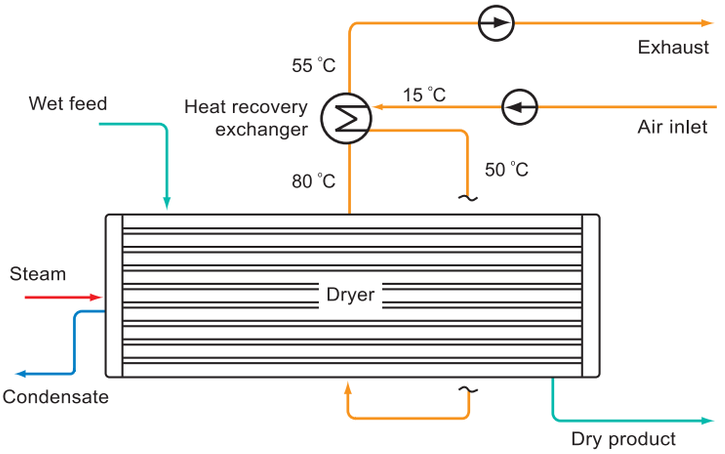


Figure 3.8 Hot Air Dryer Using Steam

The basic drying energy requirement is the latent heat needed to evaporate the water. Clearly, this depends on the amount of water being evaporated. In most cases, the product material, the carrier gas and the equipment also need to be hot. So the total energy required includes:

- Heat leaving the dryer in the exhaust flow. This includes the latent heat of the water evaporated, but the sensible heat of the hot gas can also be significant.
- Heat lost from equipment and ducting. However good the insulation, there is always some heat loss.
- Heat leaving the dryer as hot product.
- Motive power for fans and conveyors.

Common factors resulting in excessive energy use

- Excessive drying load – for example, unnecessarily wet feed material, or off-specification product that needs to be reprocessed
- Excessive airflows
- Unnecessarily hot exhaust flow
- Hot air leaks
- Poor insulation
- Excessive fan power (for example, over specified fans restricted by dampers)
- Steam system inefficiencies

Example - Heat Energy in Air Drying

A food containing 80% water is to be dried at 100°C down to moisture content of 10%. If the initial temperature of the food is 21°C, calculate the quantity of heat energy required per unit weight of the original material, for drying under atmospheric pressure. The latent heat of vaporization of water at

100°C and at standard atmospheric pressure is 2257 kJ/kg. The specific heat capacity of the food is 3.8 kJ/kg °C and of water is 4.186 kJ/kg °C. Find also the energy requirement/kg water removed.

Calculating for 1 kg food

Initial moisture = 80%

800 g moisture is associated with 200 g dry matter.

Final moisture = 10 %,

100 g moisture is associated with 900 g dry matter,

Therefore (100 x 200)/900 g = 22.2 g moisture are associated with 200 g dry matter.

1 kg of original matter must lose (800 - 22) g moisture = 778 g = 0.778 kg moisture.

Heat energy required for 1 kg original material

= heat energy to raise temperature to 100°C + latent heat to remove water
= (100 - 21) x 3.8 + 0.778 x 2257
= 300.2 + 1755.9
= 2056 kJ.

Energy/kg water removed, as 2056 kJ are required to remove 0.778 kg of water,

= 2056/0.778
= 2643 kJ.

Steam is often used to supply heat to air or to surfaces used for drying. In condensing, steam gives up its latent heat of vaporization; in drying, the substance being dried must take up latent heat of vaporization to convert its liquid into vapour, so it might be reasoned that 1 kg of steam condensing will produce 1 kg of vapour, neglecting minor losses.

3.5 Proper Selection, Operation and Maintenance of Steam Traps

The purpose of installing the steam traps is to obtain fast heating of the product and equipment by keeping the steam lines and equipment free of condensate, air and non-condensable gases. A steam trap is a valve device that discharges condensate and air from the line or piece of equipment without discharging the steam.

Functions of Steam Traps

The three important functions of steam traps are:

- To discharge condensate as soon as it is formed

- Not to allow steam to escape.
- To be capable of discharging air and other incondensable gases.

Types of Steam Traps

There are three basic types of steam trap into which all variations fall, all three are classified by International Standard ISO 6704:1982.

Thermostatic (operated by changes in fluid temperature) - The temperature of saturated steam is determined by its pressure. In the steam space, steam gives up its enthalpy of evaporation (heat), producing condensate at steam temperature. As a result of any further heat loss, the temperature of the condensate will fall. A thermostatic trap will pass condensate when this lower temperature is sensed. As steam reaches the trap, the temperature increases and the trap closes.

Mechanical (operated by changes in fluid density) - This range of steam traps operates by sensing the difference in density between steam and condensate. These steam traps include 'ball float traps' and 'inverted bucket traps'. In the 'ball float trap', the ball rises in the presence of condensate, opening a valve which passes the denser condensate. With the 'inverted bucket trap', the inverted bucket floats when steam reaches the trap and rises to shut the valve. Both are essentially 'mechanical' in their method of operation.

Thermodynamic (operated by changes in fluid dynamics) - Thermodynamic steam traps rely partly on the formation of flash steam from condensate. This group includes 'thermodynamic', 'disc', 'impulse' and 'labyrinth' steam traps.

Some of the important traps in industrial use are explained as follows:

Inverted Bucket

The inverted bucket steam trap is shown in Figure 3.9. As its name implies, the mechanism consists of an inverted bucket which is attached by a lever to a valve. An essential part of the trap is the small air vent hole in the top of the bucket. In (i) the bucket hangs down, pulling the valve off its seat. Condensate flows under the bottom of the bucket filling the body and flowing away through the outlet. In (ii) the arrival of steam causes the bucket to become buoyant, it then rises and shuts the outlet. In (iii) the trap remains shut until the steam in the bucket has condensed or bubbled through the vent hole to the top of the trap body. It will then sink, pulling the main valve off its seat. Accumulated condensate is released and the cycle is repeated.

In (ii), air reaching the trap at start-up will also give the bucket buoyancy and close the valve. The bucket vent hole is essential to allow air to escape into the top of the trap for eventual discharge through the main valve seat. The hole, and the pressure differential, is small so the trap is relatively slow at venting air. At the same time it must pass (and therefore waste) a certain amount of steam for the trap to operate once the air has cleared. A parallel air vent fitted outside the trap will reduce start-up times.

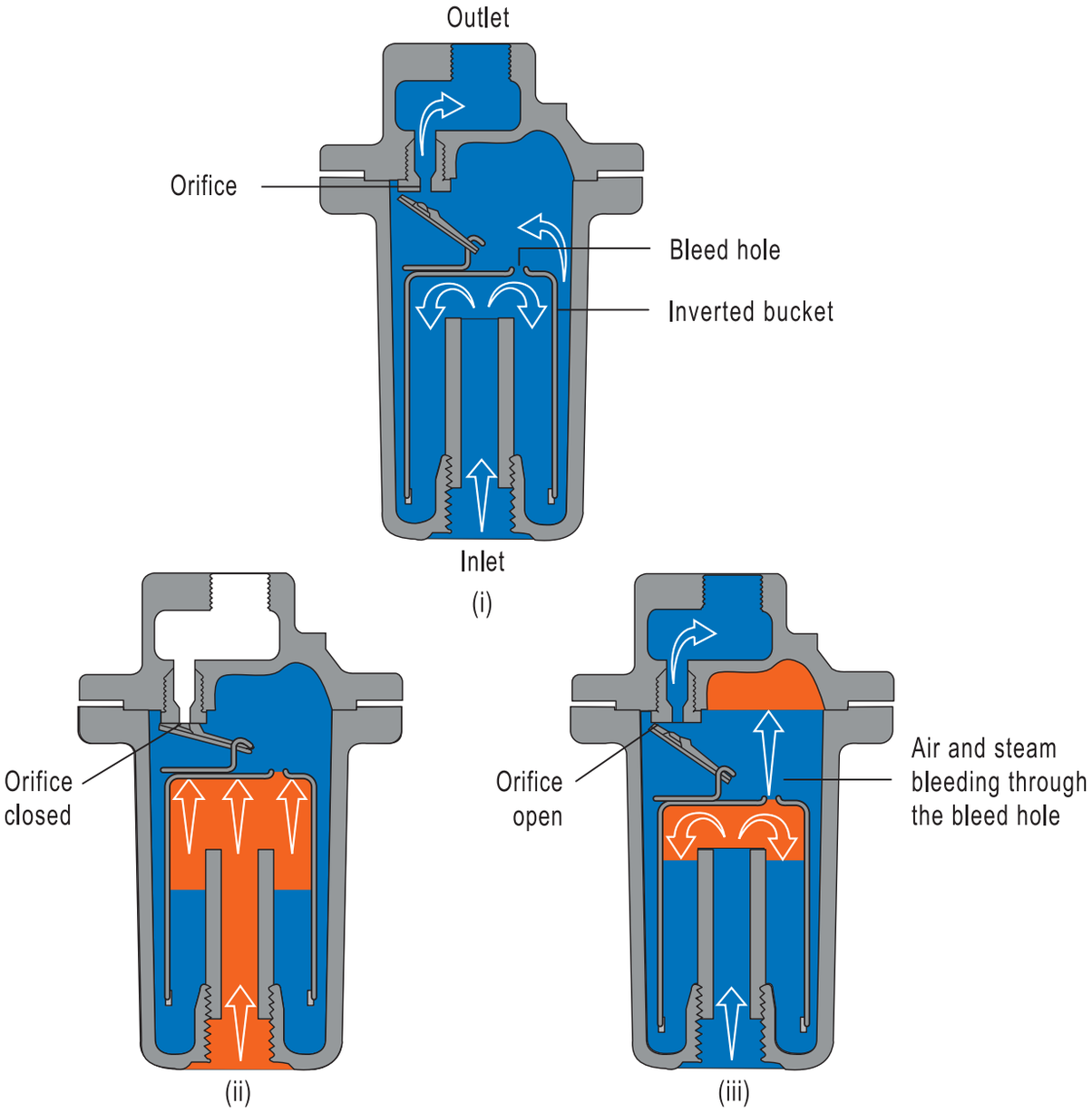


Figure 3.9 Inverted bucket trap

Advantages of the inverted bucket steam trap

- The inverted bucket steam trap can be made to withstand high pressures.
- Like a float-thermostatic steam trap, it has a good tolerance to water hammer conditions.
- Can be used on superheated steam lines with the addition of a check valve on the inlet.
- Failure mode is usually open, so it's safer on those applications that require this feature, for example turbine drains.

Disadvantages of the inverted bucket steam trap

- The small size of the hole in the top of the bucket means that this type of trap can only discharge air very slowly. The hole cannot be enlarged, as steam would pass through too quickly during normal operation.
- There should always be enough water in the trap body to act as a seal around the lip of the bucket. If the trap loses this water seal, steam can be wasted through the outlet valve. This can often happen on applications where there is a sudden drop in steam pressure, causing some of the condensate in the trap body to 'flash' into steam. The bucket loses its buoyancy and sinks, allowing live steam to pass through the trap orifice. Only if sufficient condensate reaches the trap will the water seal form again, and prevent steam wastage.

Float and Thermostatic

The ball float type trap operates by sensing the difference in density between steam and condensate. In the case of the trap shown in Figure 3.10A, condensate reaching the trap will cause the ball float to rise, lifting the valve off its seat and releasing condensate. As can be seen, the valve is always flooded and neither steam nor air will pass through it, so early traps of this kind were vented using a manually operated cock at the top of the body. Modern traps use a thermostatic air vent, as shown in Figure 3.10B. This allows the initial air to pass whilst the trap is also handling condensate.

The automatic air vent uses the same balanced pressure capsule element as a thermostatic steam trap, and is located in the steam space above the condensate level. After releasing the initial air, it remains closed until air or other non-condensable gases accumulate during normal running and cause it to open by reducing the temperature of the air/steam mixture. The thermostatic air vent offers the added benefit of significantly increasing condensate capacity on cold start-up.

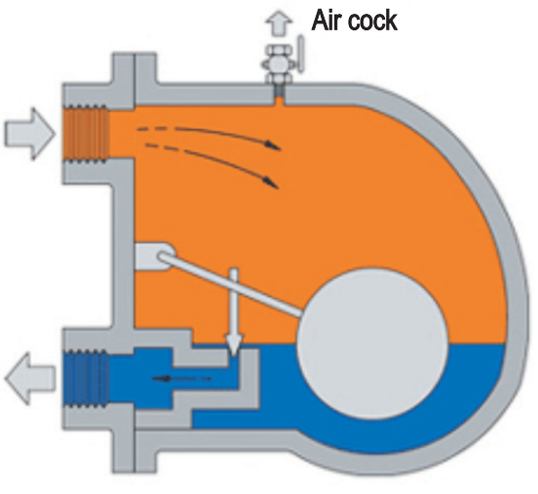


Figure 3.10A Float trap with air cock

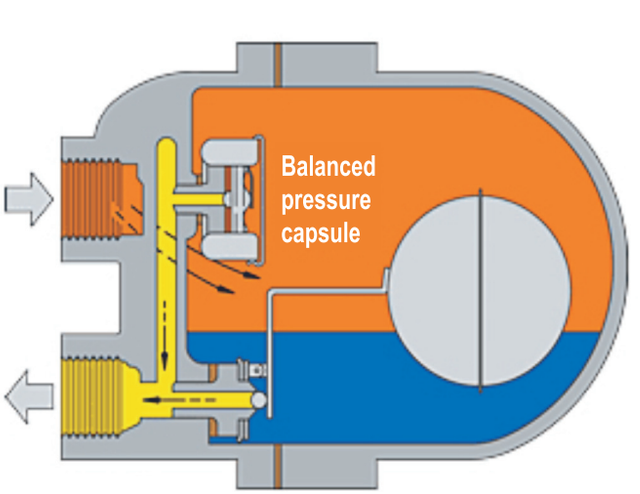


Figure 3.10B Float trap with thermostatic air vent

In the past, the thermostatic air vent was a point of weakness if water hammer was present in the system. Even the ball could be damaged if the water hammer was severe. However, in modern float traps the air vent is a compact, very robust, all stainless steel capsules, and the modern welding techniques used on the ball makes the complete float-thermostatic steam trap very robust and reliable in water hammer situations.

In many ways the float-thermostatic trap is the closest to an ideal steam trap. It will discharge condensate as soon as it is formed, regardless of changes in steam pressure.

Advantages of the float-thermostatic steam trap

- The trap continuously discharges condensate at steam temperature. This makes it the first choice for applications where the rate of heat transfer is high for the area of heating surface available.
- It is able to handle heavy or light condensate loads equally well and is not affected by wide and sudden fluctuations of pressure or flow rate.
- As long as an automatic air vent is fitted, the trap is able to discharge air freely.
- It has a large capacity for its size.
- The versions which have a steam lock release valve are the only type of trap entirely suitable for use where steam locking can occur.
- It is resistant to water hammer.

Disadvantages of the float-thermostatic steam trap

- Although less susceptible than the inverted bucket trap, the float type trap can be damaged by severe freezing and the body should be well lagged, and / or complemented with a small supplementary thermostatic drain trap, if it is to be fitted in an exposed position.
- As with all mechanical type traps, different internals are required to allow operation over varying pressure ranges. Traps operating on higher differential pressures have smaller orifices to balance the buoyancy of the float.

Thermodynamic

The thermodynamic trap is an extremely robust steam trap with a simple mode of operation. The trap operates by means of the dynamic effect of flash steam as it passes through the trap, as depicted in Figure 3.11. The only moving part is the disc above the flat face inside the control chamber or cap.

On start-up, incoming pressure raises the disc, and cool condensate plus air is immediately discharged from the inner ring, under the disc, and out through three peripheral outlets (Figure 3.11, i).

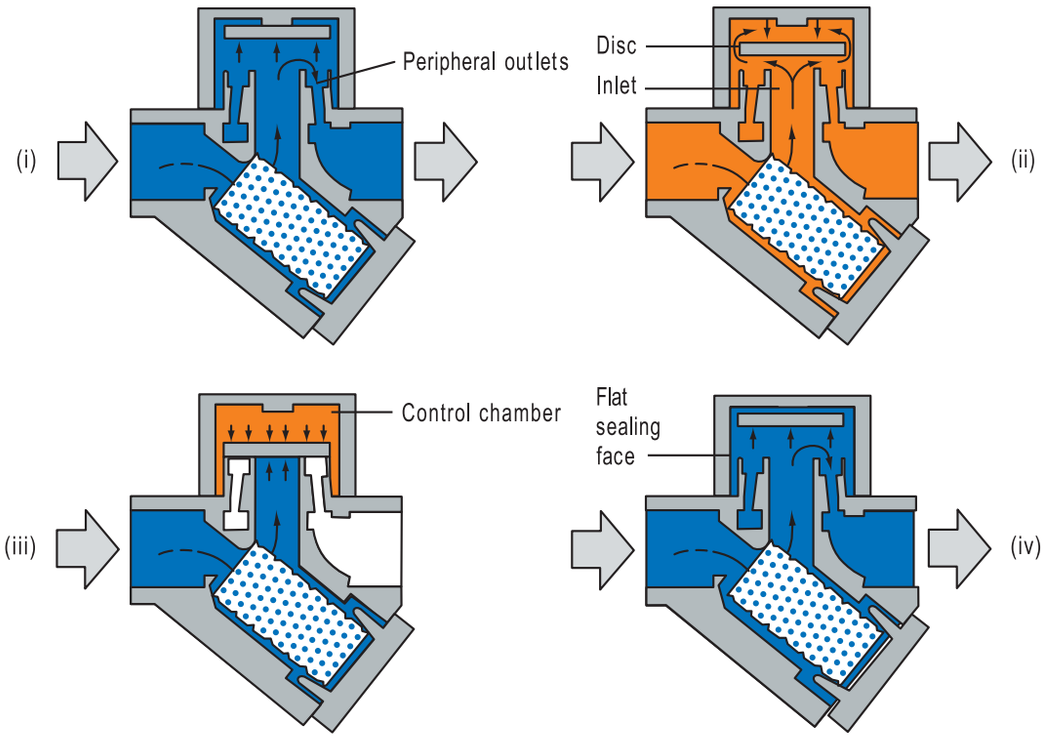


Figure 3.11 Thermodynamic Trap

Hot condensate flowing through the inlet passage into the chamber under the disc drops in pressure and releases flash steam moving at high velocity. This high velocity creates a low pressure area under the disc, drawing it towards its seat (Figure 3.11, ii).

At the same time, the flash steam pressure builds up inside the chamber above the disc, forcing it down against the incoming condensate until it seats on the inner and outer rings. At this point, the flash steam is trapped in the upper chamber, and the pressure above the disc equals the pressure being applied to the underside of the disc from the inner ring. However, the top of the disc is subject to a greater force than the underside, as it has a greater surface area.

Eventually the trapped pressure in the upper chamber falls as the flash steam condenses. The disc is raised by the now higher condensate pressure and the cycle repeats (Figure 3.11, iv).

Thermostatic

Thermal-element thermostatic traps are temperature actuated. On startup the thermal element is in a contracted position with the valve wide-open, purging condensate, air, and other non-condensable gases. As the system warms up, heat generates pressure in the thermal element, causing it to expand and throttle the flow of hot condensate through the discharge valve.

When steam follows the hot condensate into the trap, the thermal element fully expands, closing the trap. If condensate enters the trap during system operation, it cools the element, contracting it off the seat, and quickly discharging condensate (Figure 3.12).

Thermostatic traps are small, lightweight, and compact. One trap operates over extremely broad pressure and capacity ranges. Thermal elements can be selected to operate within a range of steam temperatures. In steam tracing applications it may be desirable to actually back up hot condensate in the lines to extract its thermal value.

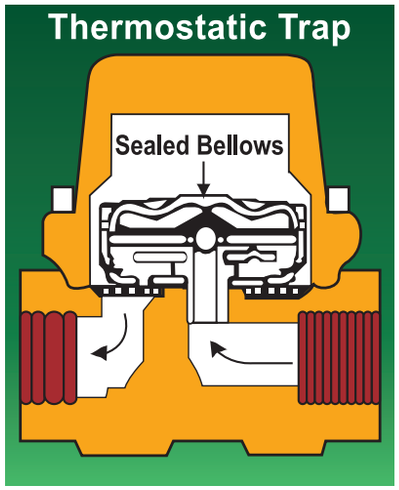


Figure 3.12 Thermostatic Trap

Bimetallic Type

Bimetallic steam traps operate on the same principle as a heating thermostat. A bimetallic strip or wafer connected to a valve bends or distorts when subjected to a change in temperature. When properly calibrated, the valve closes off against a seat when steam is present, and opens when condensate, air, and other noncondensable gases are present (Figure 3.13).

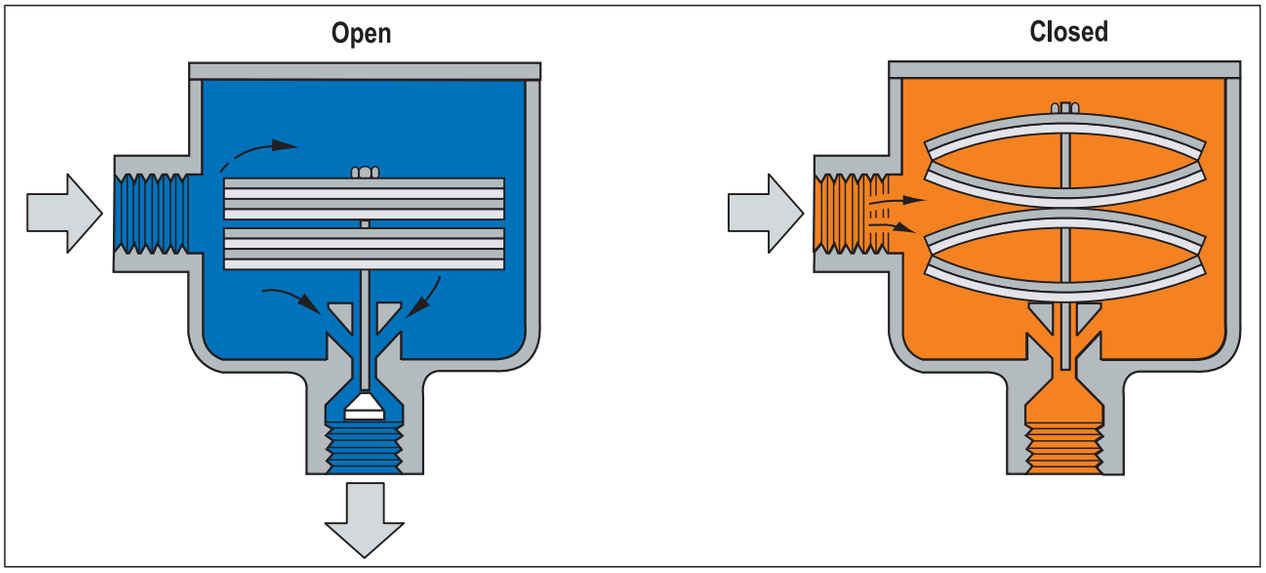


Figure 3.13 Bimetallic Trap

Advantages of the bimetallic steam trap

- Relatively small size for the condensate loads they handle
- Resistance to damage from water hammer

A disadvantage is that they must be set, generally at the plant, for a particular steam operating pressure. If the trap is used for a lower pressure, it may discharge live steam. If used at a higher steam pressure, it can back up condensate into the system.

Thermostatic traps are often considered a universal steam trap; however, they are normally not recommended for extremely high condensate requirements (over 7000 kg/hr). For light-to-moderately high condensate loads, thermostatic steam traps offer advantages in terms of initial cost, long-term energy conservation, reduced inventory, and ease in application and maintenance.

Installation of Steam Traps

In most cases, trapping problems are caused by bad installation rather than by the choice of the wrong type or faulty manufacture. To ensure a trouble-free installation, careful consideration should be given to the drain point, pipe sizing, air venting, steam locking, group trapping vs. individual trapping, dirt, water hammer, lifting of the condensate, etc.

1) Drain Point

The drain point should be so arranged that the condensate can easily flow into the trap. This is not always appreciated. For example, it is useless to provide a 15mm drain hole in the bottom of a 150 mm steam main, because most of the condensate will be carried away by the steam velocity. A proper pocket at the lowest part of the pipe line into which the condensate can drop of at least 100mm diameter is needed in such cases.

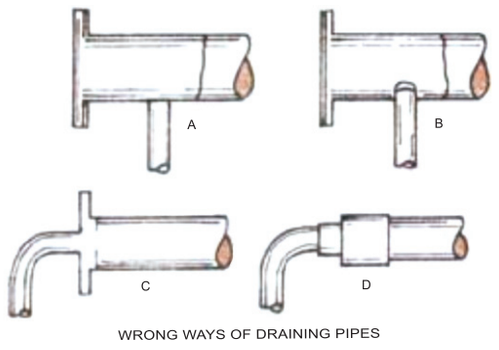


Figure 3.14 A Wrong ways of Draining Pipes

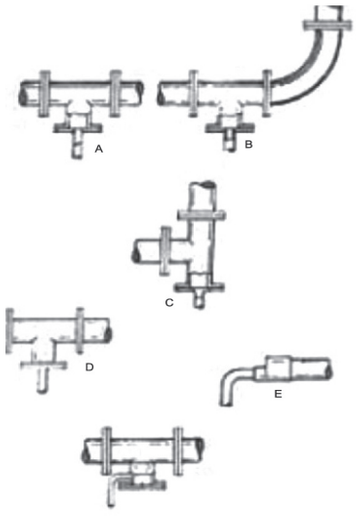


Figure 3.14 B Right ways of Draining Pipes

Figures 3.14A and 3.14B show the wrong and the correct practices in providing the drain points on the steam lines.

2) Pipe Sizing

The pipes leading to and from steam traps should be of adequate size. This is particularly important in the case of thermodynamic traps, because their correct operation can be disturbed by excessive resistance to flow in the condensate pipe work. Pipe fittings such as valves, bends and tees close to the trap will also set up excessive backpressures in certain circumstances.

3) Air Binding

When air is pumped into the trap space by the steam, the trap function ceases. Unless adequate provision is made for removing air either by way of the steam trap or a separate air vent, the plant may take a long time in warming up and may never give its full output.

4) Steam Locking

This is similar to air binding except that the trap is locked shut by steam instead of air. The typical example is a drying cylinder. It is always advisable to use a float trap provided with a steam lock release arrangement.

5) Group Trapping vs. Individual Trapping

It is tempting to try and save money by connecting several units to a common steam trap as shown in Figure 3.15A. This is known as group trapping. However, it is rarely successful, since it normally causes water-logging and loss of output.

The steam consumption of a number of units is never the same at a moment of time and therefore, the pressure in the various steam spaces will also be different. It follows that the pressure at the drain outlet of a heavily loaded unit will be less than in the case of one that is lightly or properly loaded. Now, if all these units are connected to a common steam trap, the condensate from the heavily loaded and therefore lower pressure steam space finds it difficult to reach the trap as against the higher pressure condensate produced by lightly or partly loaded unit. The only satisfactory arrangement, thus would be to drain each steam space with own trap and then connect the outlets of the various traps to the common condensate return main as shown in above Figure 3.15B.

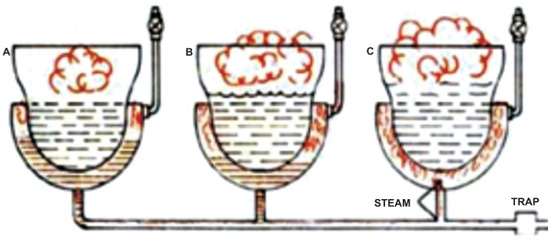


Figure 3.15A Group Trapping

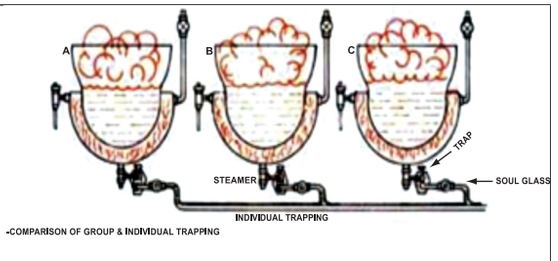


Figure 3.15B Individual Trapping

6) Dirt

Dirt is the common enemy of steam traps and the causes of many failures. New steam systems contain scale, castings, weld metal, piece of packing and jointing materials, etc. When the system has been in use for a while, the inside of the pipe work and fittings, which is exposed to corrosive condensate can get rusted. Thus, rust in the form of a fine brown powder is also likely to be present. All this dirt will be carried through the system by the steam and condensate until it reaches the steam trap. Some of it may pass through the trap into the condensate system without doing any harm, but some dirt will eventually jam the trap mechanism. It is advisable to use a strainer positioned before the steam trap to prevent dirt from passing into the system.

7) Water Hammer

A water hammer (Figure 3.16) in a steam system is caused by condensate collection in the plant or pipe work picked up by the fast moving steam and carried along with it. When this collection hits obstructions such as bends, valves, steam traps or some other pipe fittings, it is likely to cause severe damage to fittings and equipment and result in leaking pipe joints.

The problem of water hammer can be eliminated by positioning the pipes so that there is a continuous slope in the direction of flow. In case of steam mains, a slope of at least 1 m in every 100 meters is necessary, as also an adequate number of drain points every 30 to 50 meters.

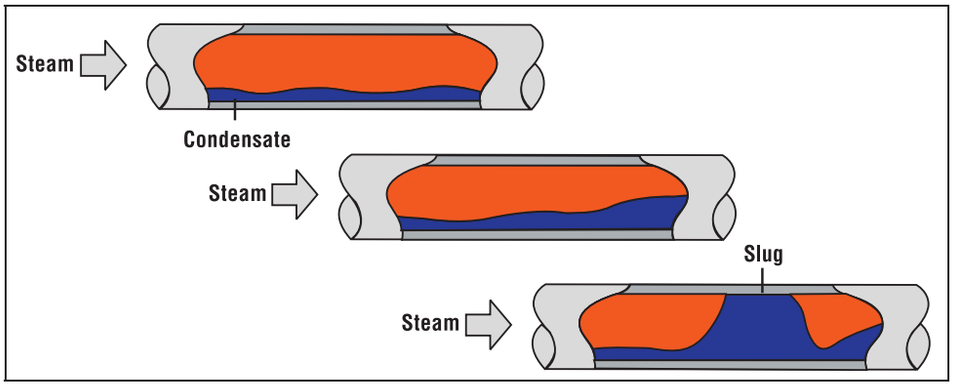


Figure 3.16 Water Hammer

8) Lifting the condensate

It is sometimes necessary to lift condensate from a steam trap to a higher level condensate return line (Figure 3.17). The condensate will rise up the lifting pipe work when the steam pressure upstream of the trap is higher than the pressure downstream of the trap.

The pressure downstream of the trap is generally called backpressure, and is made up of any pressure existing in the condensate line plus the static lift caused by condensate in the rising pipe work. The upstream pressure will vary between start-up conditions, when it is at its lowest and running conditions, when it is at its highest.

Backpressure is related to lift by using the following approximate conversion: 1 metre lift in pipe work = 1 m head static pressure or 0.1 bar backpressure.

If a head of 5 m produces a backpressure of 0.5 bar, then this reduces the differential pressure available to push condensate through the trap; although under running conditions the reduction in trap capacity is likely to be significant only where low upstream pressures are used.

In steam mains at start-up, the steam pressure is likely to be very low, and it is common for water to back-up before the trap, which can lead to water hammer in the space being drained. To alleviate this problem at start-up, a liquid expansion trap, fitted as shown in Figure 3.17, will discharge any cold condensate formed at this time to waste.

As the steam main is warmed, the condensate temperature rises, causing the liquid expansion trap to close. At the same time, the steam pressure rises, forcing the hot condensate through the 'working' drain trap to the return line.

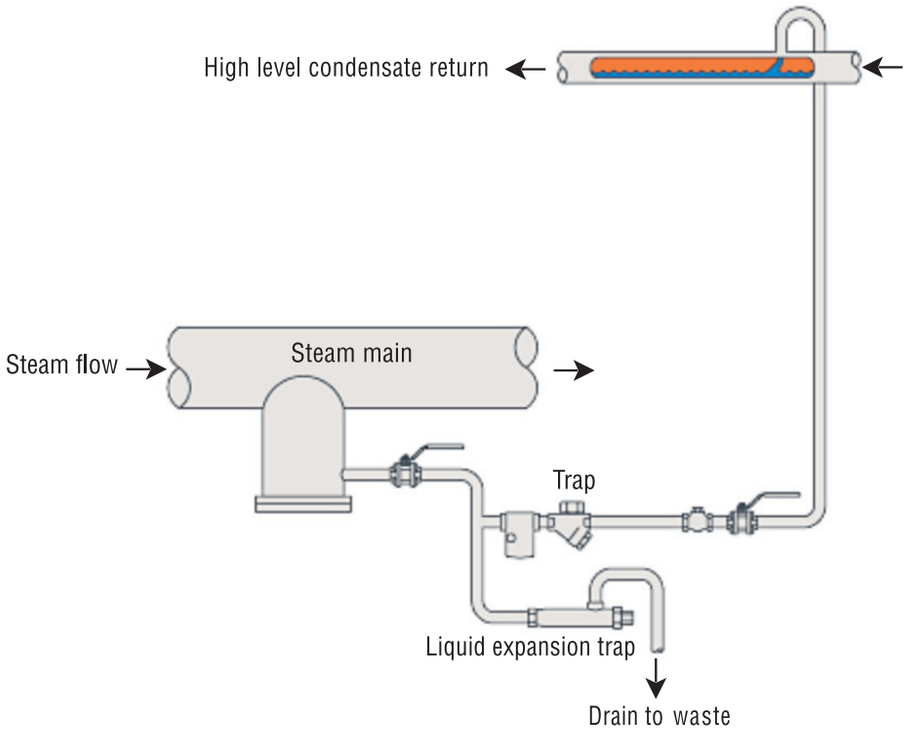


Figure 3.17 Use of a Liquid Expansion Trap

The discharge line from the trap to the overhead return line preferably discharges into the top of the main rather than simply feed to the underside, as shown in Figure 3.17. This assists operation, because although the riser is probably full of water at start-up, it sometimes contains little more than flash steam once hot condensate under pressure passes through. If the discharge line were fitted to the bottom of the return line, it would fill with condensate after each discharge and increase the tendency for water hammer and noise.

It is also recommended that a check valve be fitted after any steam trap from where condensate is lifted, preventing condensate from falling back towards the trap. The above general recommendations apply not just to traps lifting condensate from steam mains, but also to traps draining any type of process running at a constant steam pressure. Temperature controlled processes will often run with low steam pressures. Rising condensate discharge lines should be avoided at all costs, unless automatic pump-traps are used.

Maintenance of steam traps

Dirt is one of the most common causes of steam traps blowing steam. Dirt and scale are normally found in all steam pipes. Bits of jointing material are also quite common. Since steam traps are connected to the lowest parts of the system, sooner or later this foreign matter finds its way to the trap. Once some of the dirt gets logged in the valve seat, it prevents the valve from shutting down tightly thus allowing steam to escape. The valve seal should therefore be quickly cleaned, to remove this obstruction and thus prevent steam loss.

In order to ensure proper working, steam traps should be kept free of pipe-scale and dirt. The best way to prevent the scale and dirt from getting into the trap is to fit a strainer. Strainer (Figure 3.18) is a detachable, perforated or meshed screen enclosed in a metal body. It should be borne in mind that the strainer collects dirt in the course of time and will therefore need periodic cleaning. It is of course, much easier to clean a strainer than to overhaul a steam trap.

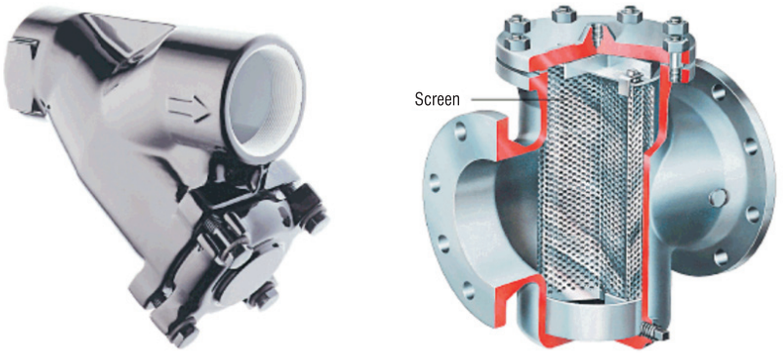


Figure 3.18 Strainers

At this point, we might mention the usefulness of a sight glass fitted just after a steam trap. Sight glasses are useful in ascertaining the proper functioning of traps and in detecting leaking steam traps. In particular, they are of considerable advantage when a number of steam traps are discharging into a common return line. If it is suspected that one of the traps is blowing steam, it can be quickly identified by looking through the sight glass.

In most industries, maintenance of steam traps is not a routine job and is neglected unless it leads to some definite trouble in the plant. In view of their importance as steam savers and to monitor plant efficiency, the steam traps require considerably more care than is given.

One may consider a periodic maintenance schedule to repair and replace defective traps in the shortest possible time, preferable during regular maintenance shut downs in preference to break down repairs.

Guide to Steam Trap Selection

Actual energy efficiency can be achieved only when

a) Selection

b) Installation and

c) Maintenance of steam traps meet the requirements for the purpose it is installed

The following Table 3.3 gives installation of suitable traps for different process applications.

Table 3.3 Selection of Steam trap

Application	Feature	Suitable trap
Steam mains	○ Open to atmosphere, small capacity ○ Frequent change in pressure ○ Low pressure - high pressure	Thermodynamic trap
Equipment ○ Reboiler ○ Heater ○ Dryer ○ Heat exchanger etc.	○ Large capacity ○ Variation in pressure and temperature is undesirable ○ Efficiency of the equipment is a problem	Mechanical trap, Bucket, Inverted bucket, Float traps
○ Tracer line ○ Instrumentation	○ Reliability with no over heating	Thermodynamic & Bimetallic traps

3.6 Performance Assessment Methods for Steam Traps

Steam trap performance assessment is basically concerned with answering the following two questions:

- Is the trap working correctly or not?
- If not, has the trap failed in the open or closed position?

Traps that fail 'open' result in a loss of steam and its energy. Where condensate is not returned, the water is lost as well. The result is significant economic loss, directly via increased boiler plant costs, and potentially indirectly, via decreased steam heating capacity.

Traps that fail 'closed' do not result in energy or water losses, but can result in significantly reduced heating capacity and/or damage to steam heating equipment.

Visual Testing

Visual testing includes traps with open discharge, sight glasses (Figure 3.19), sight checks, test tees and three way test valves. In every case, the flow or variation of flow is visually observed. This method works well with traps that cycle on/off, or dribbles on light load. On high flow or process, due to the volume of water and flash steam, this method becomes less viable. If condensate can be diverted ahead of the trap or a secondary flow can be turned off, the load on the trap will drop to zero or a very minimal amount so the visual test will allow in determining the leakage.

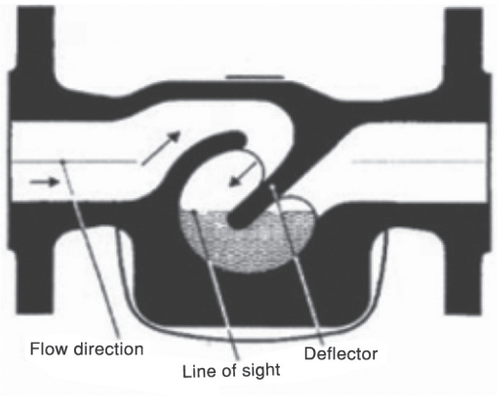


Figure 3.19 Sight Glass

Sound Testing

Sound testing includes ultrasonic leak detectors (Figure 3.20), mechanics stethoscopes, screwdriver or metal rod with a human ear against it. All these use the sound created by flow to determine the trap function like the visual method. This method works best with traps that cycle on/off or dribbles on light load. Traps which have modulating type discharge patterns are hard to check on high flows. (Examples are processes, heat exchangers, air handling coils, etc). Again by diverting condensate flow ahead of the trap or shutting off a secondary flow as mentioned under visual testing, the noise level will drop to zero or a very low level if the trap is operating correctly. If the trap continues to flow heavily after diversion it would be leaking or blowing through.

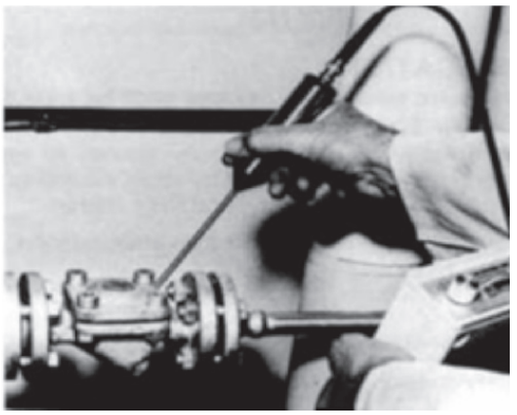


Figure 3.20 Ultrasonic Testing

Temperature Testing

Temperature testing includes infrared guns (Figure 3.21), surface pyrometers, temperature tapes, and temperature crayons. Typically they are used to gauge the discharge temperature on the outlet side of the trap. In the case of temperature tapes or crayon, they are set for a predetermined temperature and they indicate when temperature exceeds that level. Infrared guns and surface pyrometer can detect temperatures on both sides of the trap. Both the infrared and surface pyrometers require bare pipe and

a clean surface to achieve a reasonable reading. The temperature reading will typically be lower than actual internal pipe temperature due to the fact that steel does have some heat flow resistance. Scale on the inside of the pipe can also affect the heat transfer. Some of the more expensive infrared guns can compensate for wall thickness and material differences. Blocked or turned off traps can easily be detected by infrared guns and surface pyrometers, as they will show low or cold temperatures. They could also pick up traps which may be undersized or backing up large amounts of condensate by detecting low temperature readings.

Figure 3.21 Infra Red Testing

3.7 Efficient Steam Utilisation and Energy Saving Opportunities

1. Monitoring Steam Traps

For testing a steam trap, there should be an isolating valve provided in the downstream of the trap and a test valve shall be provided in the trap discharge. When the test valve is opened, the following points have to be observed:

Condensate discharge-Inverted bucket and thermodynamic disc traps should have intermittent condensate discharge. Float and thermostatic traps should have a continuous condensate discharge. Thermostatic traps can have either continuous or intermittent discharge depending upon the load. If inverted bucket traps are used for extremely small load, it will have a continuous condensate discharge.

Flash steam-This shall not be mistaken for a steam leak through the trap. The users sometimes get confused between a flash steam and leaking steam. The flash steam and the leaking steam can be approximately identified as follows:

- If steam blows out continuously in a blue stream, it is a leaking steam.
- If a steam floats out intermittently in a whitish cloud, it is a flash steam.

2. Continuous steam blow and no flow indicate, there is a problem in the trap.

Whenever a trap fails to operate and the reasons are not readily apparent, the discharge from the trap should be observed. A step-by-step analysis has to be carried out mainly with reference to lack of discharge from the trap, steam loss, continuous flow, sluggish heating, to find out whether it is a system problem or the mechanical problem in the steam trap.

3. Avoiding Steam Leakages

Steam leakage is a visible indicator of waste and must be avoided. It has been estimated that a 3 mm diameter hole on a pipeline carrying 7kg/cm^2 steam would waste 33 KL of fuel oil per year. Steam leaks on high-pressure mains are prohibitively costlier than on low pressure mains. Any steam leakage must be quickly attended to. In fact, the plant should consider a regular surveillance programme for identifying leaks at pipelines, valves, flanges and joints. Indeed, by plugging all leakages, one may be surprised at the extent of fuel savings, which may reach up to 5% of the steam consumption in a small or medium scale industry or even higher in installations having several process departments.

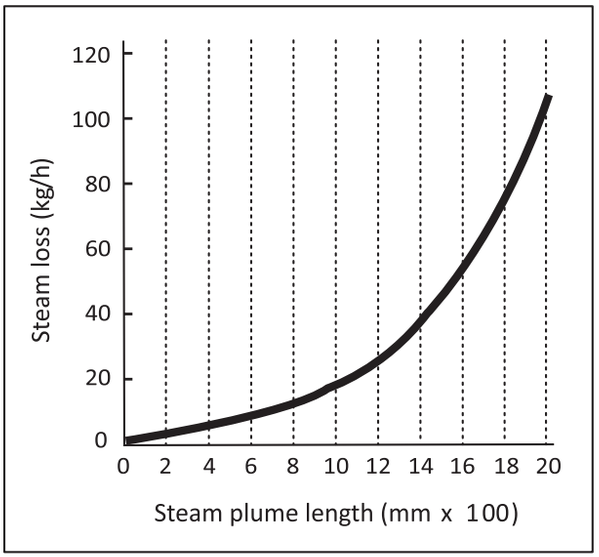


Figure 3.22 Steam Loss vs Plume Length

To avoid leaks it may be worthwhile considering replacement of the flanged joints which are rarely opened in old plants by welded joints. Figure 3.22 provides a quick estimate for steam leakage based on plume length.

Example

- Plume Length = 700 mm
- Steam loss = 10 kg/h

The following Table 3.4 highlights the significance of loss through steam leaks.

Table: 3.4 Steam Losses due to Leakage

S. No.	Dia. of Leak (in mm)	Annual Steam Loss (Tons/year)	
		At 3.5 kg/cm^2	At 7.0 kg/cm^2
1	1.5	29.1	47.3
2	3.0	116.4	192.7
3	4.5	232.7	432.7
4	6.0	465.4	767.3

4. Providing Dry Steam for Process

The best steam for industrial process heating is the dry saturated steam. Wet steam reduces total heat in the steam. Also water forms a wet film on heat transfer and overloads traps and condensate equipment. Super heated steam is not desirable for process heating because it gives up heat at a rate slower than the condensation heat transfer of saturated steam.

It must be remembered that a boiler without a super heater cannot deliver perfectly dry saturated steam. At best, it can deliver only 95% dry steam. The dryness fraction of steam depends on various factors, such as the level of water to be a part of the steam. Indeed, even as simple a thing as improper boiler water treatment can become a cause for wet steam.

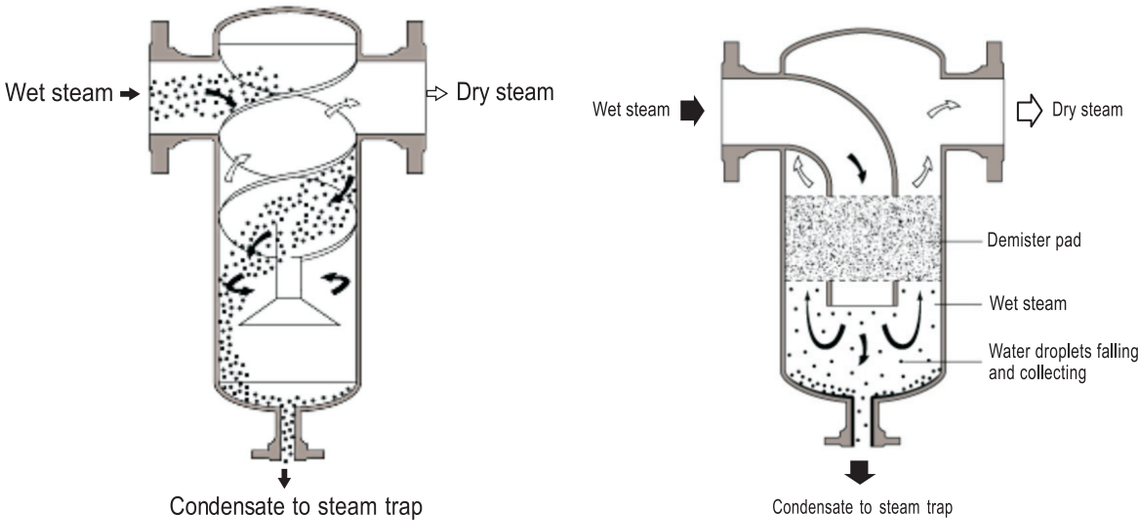
As steam flows through the pipelines, it undergoes progressive condensation due to the loss of heat to the colder surroundings; the extent of the condensation depends on the effectiveness of the lagging. For example, with poor lagging, the steam can become excessively wet.

Since dry saturated steam is required for process equipment, due attention must be paid to the boiler operation and lagging of the pipelines.

Wet steam can reduce plant productivity and product quality, and can cause damage to most items of plant and equipment. Whilst careful drainage and trapping can remove most of the water, it will not deal with the water droplets suspended in the steam. To remove these suspended water droplets, separators are installed in steam pipelines.

The steam produced in a boiler designed to generate saturated steam is inherently wet. Although the dryness fraction will vary according to the type of boiler, most shell type steam boilers will produce steam with a dryness fraction of between 95 and 98%. The water content of the steam produced by the boiler is further increased if priming and carryover occur.

A steam separator (Refer Figure 3.23) may be installed on the steam main as well as on the branch lines to reduce wetness in steam and improve the quality of the steam going to the units. By change of direction of steam, steam separators causes the entrained water particles to be separated out and delivered to a point where they can be drained away as condensate through a conventional steam trap. A few types of separators are illustrated in the Figure 3.23 below



A cyclonic type separator **A coalescence type separator**

Figure 3.23 Steam Separators

5. Utilising Steam at the Lowest Acceptable Pressure for the Process

A study of the steam tables would indicate that the latent heat in steam reduces as the steam pressure increases. It is only the latent heat of steam, which takes part in the heating process when applied to an indirect heating system. Thus, it is important that its value be kept as high as possible. This can only be achieved if we go in for lower steam pressures. As a guide, the steam should always be generated and distributed at the highest possible pressure, but utilized at as low a pressure as possible since it then has higher latent heat.

However, it may also be seen from the steam tables that the lower the steam pressure, the lower will be its temperature. Since temperature is the driving force for the transfer of heat at lower steam pressures, the rate of heat transfer will be slower and the processing time greater. In equipment where fixed losses are high (e.g. big drying cylinders), there may even be an increase in steam consumption at lower pressures due to increased processing time. There are, however, several equipments in certain industries where one can profitably go in for lower pressures and realize economy in steam consumption without materially affecting production time.

Therefore, there is a limit to the reduction of steam pressure. Depending on the equipment design, the lowest possible steam pressure with which the equipment can work should be selected without sacrificing either on production time or on steam consumption.

6. Proper Utilization of Directly Injected Steam

The heating of a liquid by direct injection of steam is often desirable. The equipment required is relatively simple, cheap and easy to maintain. No condensate recovery system is necessary. The heating is quick, and the sensible heat of the steam is also used up along with the latent heat, making the process thermally efficient. In processes where dilution is not a problem, heating is done by blowing steam into the liquid (i.e) direct steam injection is applied. If the dilution of the tank contents and agitation are not acceptable in the process (i.e) direct steam agitation are not acceptable, indirect steam heating is the only answer.

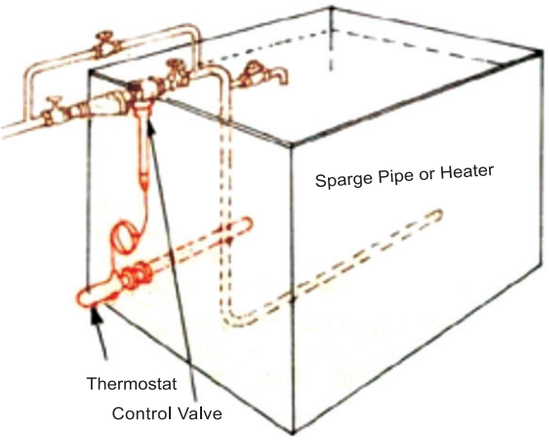


Figure 3.24 Temperature Control for Directly Injected Steam

Ideally, the injected steam should be condensed completely as the bubbles rise through the liquid. This is possible only if the inlet steam pressures are kept very low—around 0.5 kg/cm^2 –and certainly not exceeding 1 kg/cm^2. If pressures are high, the velocity of the steam bubbles will also be high and they will not get sufficient time to condense before they reach the surface. Figure 3.24 shows a recommended arrangement for direct injection of steam.

A large number of small diameter holes (2 to 5mm), facing downwards, should be drilled on the separate pipe. This will help in dissipating the velocity of bubbles in the liquid. A thermostatic control of steam admitted is highly desirable

7. Minimising Heat Transfer Barriers

The metal wall may not be the only barrier in a heat transfer process. There is likely to be a film of air, condensate and scale on the steam side. On the product side there may also be baked-on product or scale, and a stagnant film of product.

Agitation of the product may eliminate the effect of the stagnant film, whilst regular cleaning on the product side should reduce the scale.

Regular cleaning of the surface on the steam side may also increase the rate of heat transfer by reducing the thickness of any layer of scale; however, this may not always be possible. This layer may also be reduced by careful attention to the correct operation of the boiler, and the removal of water droplets carrying impurities from the boiler.

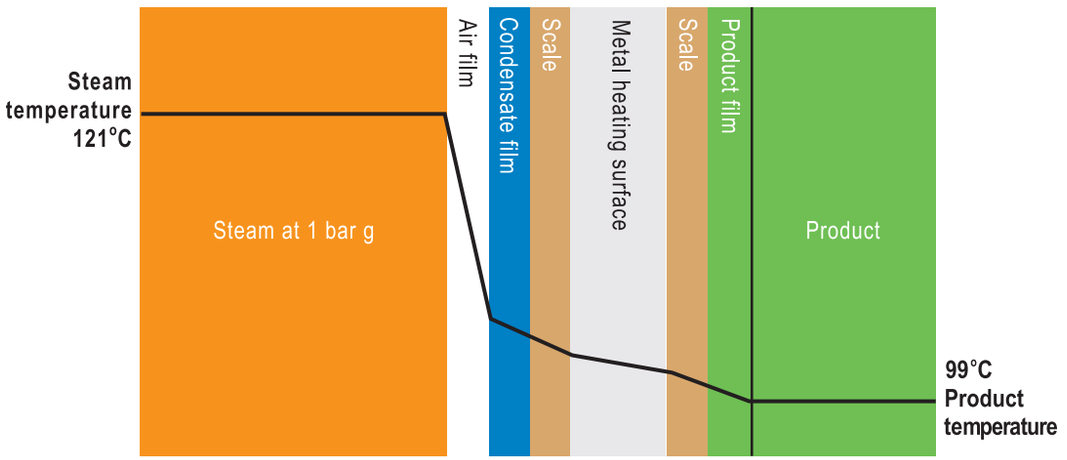


Figure 3.25 Heat Transfer Barriers

The elimination of the condensate film is not quite as simple. As the steam condenses to give up its enthalpy of evaporation, droplets of water may form on the heat transfer surface. These may merge together to form a continuous film of condensate. The condensate film may be between 100 and 150 times more resistant to heat transfer than a steel heating surface, and 500 to 600 times more resistant than copper.

As air is such a good insulator, it provides even more resistance to heat transfer. Air may be between 1500 and 3000 times more resistant to heat flow than steel, and 8000 to 16000 more resistant than copper. This means that a film of air only 0.025 mm thick may resist as much heat transfer as a wall of copper 400 mm thick. Of course all of these comparative relationships depend on the temperature profiles across each layer.

Figure 3.25 illustrates the effect this combination of layers has on the heat transfer process. These barriers to heat transfer not only increase the thickness of the entire conductive layer, but also greatly reduce the mean thermal conductivity of the layer.

The more resistant the layer to heat flow, the larger the temperature gradient is likely to be. This means that to achieve the same desired product temperature, the steam pressure may need to be significantly higher.

The presence of air and water films on the heat transfer surfaces of either process or space heating applications is not unusual. It occurs in all steam heated process units to some degree.

To achieve the desired product output and minimise the cost of process steam operations, a high heating performance may be maintained by reducing the thickness of the films on the condensing surface. In practice, air will usually have the most significant effect on heat transfer efficiency, and its removal from the supply steam will increase heating performance.

8. Proper Air Venting

Proper air venting is required because of the effect air has upon heat transfer. A layer of air only 1 mm thick can offer the same resistance to heat as a layer of water 25 µm thick, a layer of iron 2 mm thick or a layer of copper 15 mm thick. It is very important therefore to remove air from any steam system.

9. Condensate Recovery

The steam condenses after giving off its latent heat in the heating coil or the jacket of the process equipment. A sizable portion (about 25%) of the total heat in the steam leaves the process equipment as hot water. Figure 3.26 compares the amount of energy in a kilogram of steam and condensate at the same pressure. The percentage of energy in condensate to that in steam can vary from 18% at 1 bar g to 30% at 14 bar g; clearly the liquid condensate is worth reclaiming.

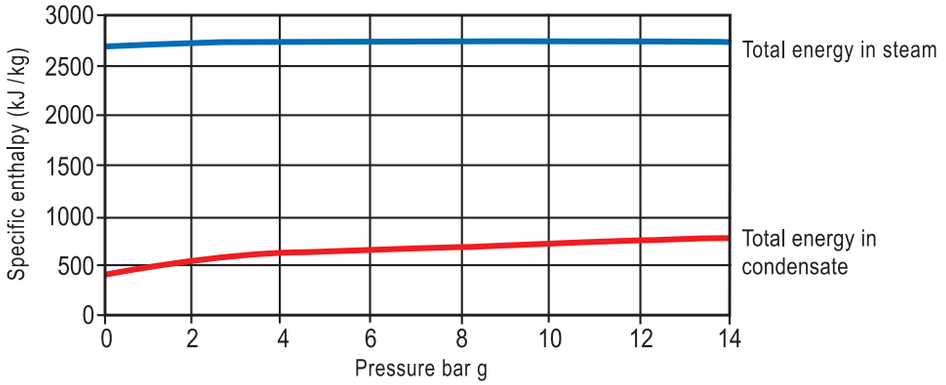


Figure: 3.26 Heat Content of Steam and Condensate (at the same pressure)

If this water is returned to the boiler house, it will reduce the fuel requirements of the boiler. For every 6 ^{0}C rise in the feed water temperature, there will be approximately 1% saving of fuel in the boiler.

Benefits of condensate recovery

Financial reasons

Condensate is a valuable resource and even the recovery of small quantities is often economically justifiable. The discharge from a single steam trap is often worth recovering.

Un-recovered condensate must be replaced in the boiler house by cold make-up water with additional costs of water treatment and fuel to heat the water from a lower temperature.

Water charges

Any condensate not returned needs to be replaced by make-up water, incurring further water charges from the local water supplier.

Effluent restrictions

High temperature of effluent is detrimental to the environment and may damage to pipes. Condensate above this temperature must be cooled before it is discharged, which may incur extra energy costs.

Maximising boiler output

Colder boiler feed water will reduce the steaming rate of the boiler. The lower the feed water temperature, the more heat, and thus fuel needed to heat the water.

Boiler feed water quality

Condensate is distilled water, which contains almost no total dissolved solids (TDS). Boilers need to be blown down to reduce their concentration of dissolved solids in the boiler water. Returning more condensate to the feed tank reduces the need for blow down and thus reduces the energy lost from the boiler.

Summary of reasons for condensate recovery:

- Water charges are reduced.
- Effluent charges and possible cooling costs are reduced.
- Fuel costs are reduced.
- More steam can be produced from the boiler.
- Boiler blow down is reduced - less energy is lost from the boiler.
- Chemical treatment of raw make-up water is reduced.

Condensate Pumping

The condensate generated from the steam consuming equipment or process widely distributed in the plant, is required to be pumped back to the condensate tank in the boiler house. The limitation with hot condensate, which is generally close to 100^{o}C, is cavitation of the pump and pump impeller.

Centrifugal pumps are prone to cavitation problem, if Net Positive Suction Head (NPSH) available is less than design NPSH required. To avoid the problem there are two alternative solutions:

- Increase the NPSH available.
- Use a pressure powered (steam driven) pump instead of a centrifugal pump

Increase the NPSH available

This can be achieved by:

- **increasing the difference in the pressure** in the receiver and the condensate pressure, and/or
- **increasing the static head in suction side** by lifting the receiver or lowering the pump, and/or
- **increasing the piping dimensions** for minimizing the friction loss in the suction pipe

Use a Steam Driven Pump

A pressure powered pump (Figure 3.27) uses steam pressure to push the condensate from the receiver back to the boiler house. In principle it consists of a receiver which receives condensate from different process/equipment. Once the condensate reaches a set level, the steam valve is opened and the steam pressure pushes the condensate to the boiler room. The operation is cyclic in nature. The advantage is pumping of condensate without losing much heat in the form of flash steam without any cavitation problems.

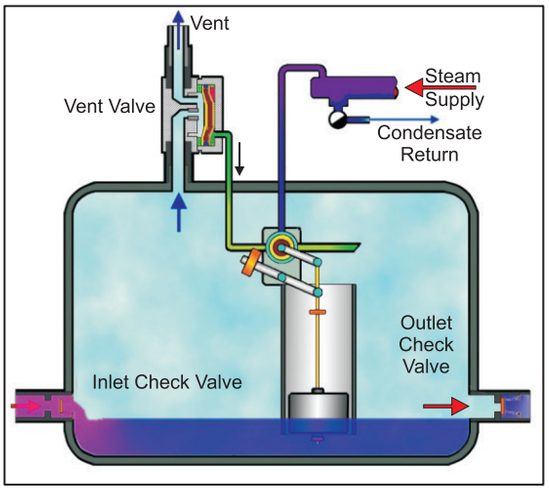


Figure 3.27 Pressure Powered Pump

10. Insulation of Steam Pipelines and Hot Process Equipments

Steam lines including flanges and valves should be insulated to prevent heat loss. The recommended thickness of insulation will mainly depend on surface temperature desired after insulation. The energy and cost savings will depend on the size of the pipe (diameter and length of run), the temperature of steam and the surroundings, heat transfer co-efficient and the number of hours of operation of the plant.

Note: Reference calculation is given in Chapter-5 “Insulation and Refractories”.

11. Flash Steam Recovery

Flash steam is produced when condensate at a high pressure is released to a lower pressure and can be used for low pressure heating.

The higher the steam pressure and lower the flash steam pressure the greater the quantity of flash steam that can be generated. In many cases, flash steam from high pressure equipments is made use of directly on the low pressure equipments to reduce use of steam through pressure reducing valves.

The flash steam quantity can be calculated by the following formula with the help of a steam table:

$$\text{Flash steam available \%} = \frac{S_1 - S_2}{L_2}$$

Where,

S_1 is the sensible heat of higher pressure condensate.

S_2 is the sensible heat of the steam at lower pressure (at which it has been flashed).

L_2 is the latent heat of flash steam (at lower pressure).

Example: Calculating the amount of flash steam from condensate

Hot condensate at 7 bar g has a heat content of about 721 kJ/kg. When it is released to atmospheric pressure (0 bar g), each kilogram of water can only retain about 419 kJ of heat. The excess energy in each kilogram of the condensate is therefore 721 – 419 = 302 kJ. This excess energy is available to evaporate some of the condensate into steam, the amount evaporated being determined by the proportion of excess heat to the amount of heat required to evaporate water at the lower pressure, which in this example, is the enthalpy of evaporation at atmospheric pressure, 2258 kJ/kg.

$$\text{Therefore, in this example, the percentage of flash steam evaporated} = \frac{302}{2258} \times 100\%$$

Flash steam evaporated = 13.4%

Example: Proportion of flash steam using Figure 3.28:

Pressure on the trap	= 4 bar g
Flash steam pressure	= 0 bar g
% Flash steam	= 10%

The amount of flash steam in the pipe is the most important factor when sizing trap discharge lines.

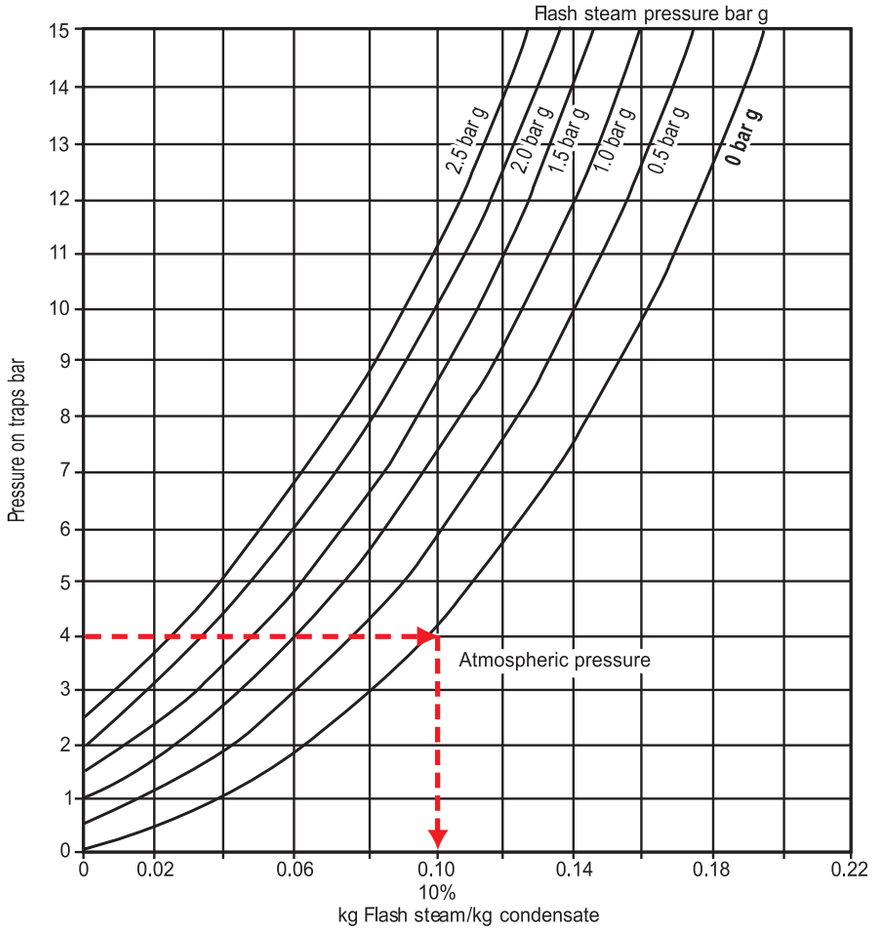


Figure 3.28 Quantity of Flash Steam Graph

Flash steam can be used on low pressure applications like direct injection and can replace an equal quantity of live steam that would be otherwise required. The demand for flash steam should exceed its supply, so that there is no build up of pressure in the flash vessel and the consequent loss of steam through the safety valve. Generally, the simplest method of using flash steam is to flash from a machine/equipment at a higher pressure to a machine/equipment at a lower pressure, thereby augmenting steam supply to the low pressure equipment.

In general, a flash system should run at the lowest possible pressure so that the maximum amount of flash is available and the backpressure on the high pressure systems is kept as low as possible.

Flash steam from the condensate can be separated in an equipment called the ‘flash vessel’. This is a vertical vessel as shown in the Figure 3.29. The diameter of the vessel is such that a considerable drop in velocity allows the condensate to fall to the bottom of the vessel from where it is drained out by a steam trap preferably a float trap. Flash steam itself rises to leave the vessel at the top. The height of the vessel should be sufficient enough to avoid water being carried over in the flash steam.

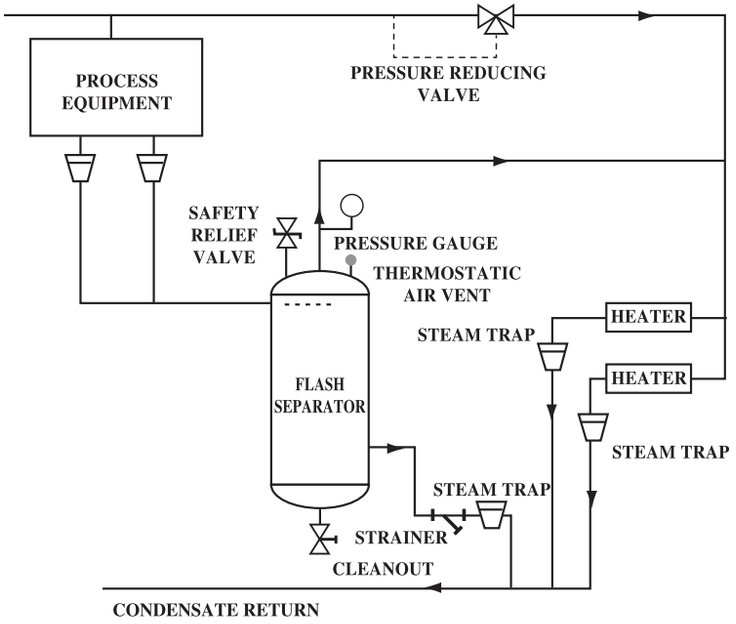


Figure 3.29 Flash Steam Recovery

The condensate from the traps (A) along with some flash steam generated passes through vessel (B). The flash steam is let out through (C) and the residual condensate from (B) goes out through the steam trap (D). The flash vessel is usually fitted with a 'pressure gauge' to know the quality of flash steam leaving the vessel. A 'safety valve' is also provided to vent out the steam in case of high pressure build up in the vessel.

12. Pipe Redundancy

All redundant (piping which are no longer needed) pipelines must be eliminated, which could be, at times, up to 10-15 % of total length. This would reduce steam distribution losses significantly.

13. Reducing the Work to be done by Steam

The equipments should be supplied with steam as dry as possible. The plant should be made efficient. For example, if any product is to be dried such as in a laundry, a press could be used to squeeze as much water as possible before being heated up in a dryer using steam.

Therefore, to take care of the above factors, automatic draining is essential and can be achieved by steam traps. The trap must drain condensate, to avoid water hammer, thermal shock and reduction in heat transfer area. The trap should also evacuate air and other non-condensable gases, as they reduce the heat transfer efficiency and also corrode the equipment. Thus, a steam trap is an automatic valve that permits passage of condensate, air and other non-condensable gases from steam mains and steam using equipment, while preventing the loss of steam in the distribution system or equipment.

The energy saving is affected by following measures:

- Reduction in operating hours
- Reduction in steam quantity required per hour
- Use of more efficient technology
- Minimizing wastage.

When the steam reaches the place where its heat is required, it must be ensured that the steam has no more work to do than is absolutely necessary. Air-heater batteries, for example, which provide hot air for drying, will use the same amount of steam whether the plant is fully or partly loaded. So, if the plant is running only at 50 per cent load, it is wasting twice as much steam (or twice as much fuel) than necessary.

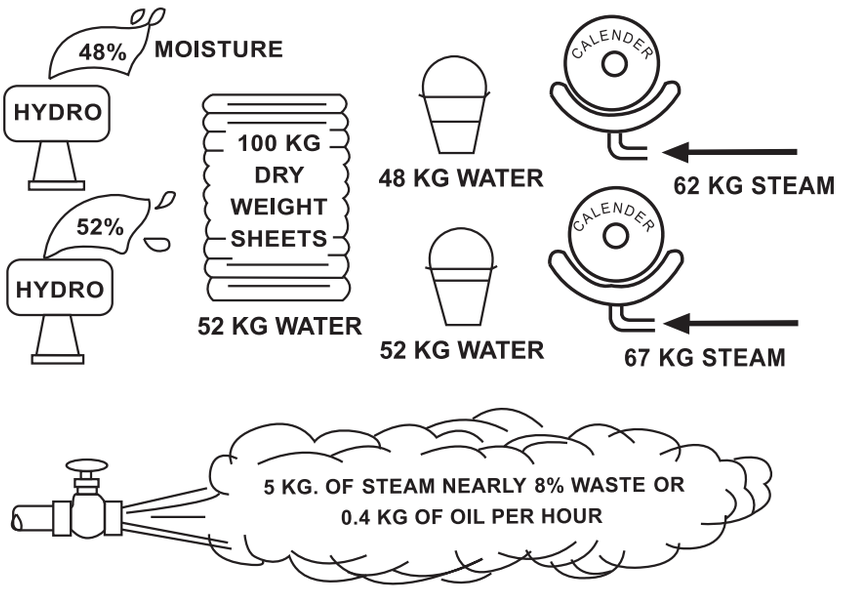


Figure 3.30 Steam Wastage Due to Insufficient Mechanical Drying

Always use the most economical way to removing the bulk of water from the wet material. Steam can then be used to complete the process. For this reason, hydro-extractors, spin dryers, squeeze or calendar rolls, presses, etc. are initially used in many drying processes to remove the mass of water. The efficiency with which this operation is carried out is most important. For example, in a laundry for finishing sheets (100 kg/hr. dry weight), the normal moisture content of the sheets as they leave the hydro extractor is 48% by weight.

Thus, the steam heated iron has to evaporate nearly 48 kg of water. This requires 62 kg of steam. If, due to inefficient drying in the hydro-extractor, the steam arrive at the iron with 52% moisture content i.e. 52 kg of water has to be evaporated, requiring about 67 kg of steam. So, for the same quantity of finished product, the steam consumption increases by 8 per cent. This is illustrated in Figure 3.30.

Solved Example:

In a crude distillation unit of a refinery, 50 Metric Tonne/hr of crude is heated using saturated steam in a heat exchanger from 35°C to 85°C. Plant is operating for 8000 hrs/annum. Consider specific heat of the crude as 0.631 kcal/kg°C. The plant has two steam headers operating at 3 bar and 8 bar respectively, passing nearby the heat exchanger. Cost of steam is same for both 3 bar and 8 bar @ Rs.4.50/kg.

As an Energy Manager, which of the following options will you recommend to the unit based on the annual cost of steam?

a) Utilising 3 bar steam
b) Utilising 8 bar steam

Given: Data from steam table:

Steam Pressure, bar	Enthalpy kcal/kg		
	Water	Evaporation	Steam
3.0	133	517	650
8.0	171	489	660

Ans:

Heat gain in crude = m * Cp * del T
Heat gain in crude = 50 * 1000 * 0.631 * (85 – 35)
Heat gain in crude = 1577500 kcal/hr
Heat gain in crude = heat loss in steam
Heat loss in steam = mass of the steam * latent heat of steam

Option A:

3 bar pressure steam having 517 kcal/kg of latent heat
Mass of the steam = 1577500 / 517
Mass of the steam = 3051 kg / hr
Cost of steam expenditure = 3051 * 8000 * 4.5
Cost of steam expenditure = 10.98 Cr

Option B:

8 bar pressure steam having 489 kcal/kg of latent heat
Mass of the steam = 1577500 / 489
Mass of the steam = 3225 kg / hr
Cost of steam expenditure = 3226 * 8000 * 4.5
Cost of steam expenditure = 11.61 Cr

Recommendation: Option A is recommended as it is found to be economical (Steam @ 3 bar pressure) since the expenditure per annum is less when compared to Option B.

QUESTIONS	
Objective Type Questions	
1.	The heat which is required to change the phase from water at 100 ^{0}C to saturated steam is called a) Latent Heat b) Sensible Heat c) Super Heat d) Specific Heat
2.	Steam mains should be run with a falling slope of … in the direction of steam flow for effective line condensate drainage a) 50 mm in 30 meters b) 125 mm in 30 meters c) 250 mm in 30 meters d) 350 mm in 30 meters
3.	The major cause for steam trap blowing steam is a) dirt b) too much condensate c) too much steam d) too much air
4.	The normal velocities encountered in pipes for superheated steam is a) 50-70 m/s b) 30-40 m/s c) 20-25 m/s d) 15-20 m/s
5.	Ideal trap for steam mains is a) thermodynamic b) float c) inverted bucket d) bimetallic
6.	The best quality of steam for industrial process heating is a) Dry saturated b) Super heated c) Wet Steam d) High pressure steam
7.	1% fuel can be saved in the boiler fuel consumption, if feed water temperature is increased by a) 6 ^{o}C b) 10 ^{o}C c)12 ^{o}C d) 22 ^{o}C
8.	Lagging of steam pipes is done to prevent a) Heat loss b) Steam leaks c) High pressures d) Pipe damages
9.	The interval for drain points in a normal steam main line should not be more than: a) 50-70 meter b) 30-50 meter c) 80-100 meter d) 20-30 meter
10.	Which of the steam traps operates on the principle of difference in density between steam and condensate a) thermostatic b) inverted bucket c) bi-metallic d) none of the above
Short Type Questions	
S-1	Explain why steam is used as a heat transfer medium?
S-2	A paint dryer requires about 85.64 m^3/min of 93.3 ^{o}C air, which is heated in a steam-coil unit. How many kg of 3.5 kg/cm^2 steam does this unit require per hour? The density of air is 1.2 kg/m^3. Specific heat of air is 0.24 kcal/kg ^{o}C.
S-3	Explain why low-pressure steam is preferred for indirect heating

S-4	Explain, why de-superheating is done after pressure reduction
S-5	What happens to steam properties such as saturation temperature, enthalpy of evaporation, latent heat of steam, enthalpy of steam and specific volume of steam if the steam pressure is increased?
Long Type Questions	
L-1	Discuss advantages of condensate and flash steam recovery in steam systems.
L-2	(i) Draw the steam phase diagram in a coordinate system with temperature as (y-axis) and enthalpy as (x-axis) (ii) Explain the major regions of the diagram.

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Chapter 4

Furnaces

Learning Objectives

In this chapter you will learn about

- Types and classification
- Methods of evaluating performance
- Various losses in the furnace
- Instruments required for monitoring performance
- Energy saving opportunities

4. FURNACES

Classification, Performance Evaluation of a Typical Furnace, Induction furnace, General Fuel Economy Measures in Furnaces, Excess Air, Heat Distribution, Temperature Control, Draft Control, Waste Heat Recovery

A furnace is an equipment to melt metals for casting or heat materials for change of shape (Rolling, forging etc) or change of properties (heat treatment).

4.1 Types and Classification of Different Furnaces

Based on the method of generating heat, furnaces are broadly classified into two types namely combustion type (using fuels) and electric type. The combustion type furnace can be broadly classified as oil fired, coal fired or gas fired, depending upon the kind of combustion.

- Based on the mode of charging of material furnaces can be classified as (i) Intermittent or Batch type furnace or Periodical furnace and (ii) Continuous furnace.
- Based on mode of waste heat recovery as recuperative and regenerative furnaces.
- Another type of furnace classification is made based on mode of heat transfer, mode of charging and mode of heat recovery as shown in the Figure 4.1 below.

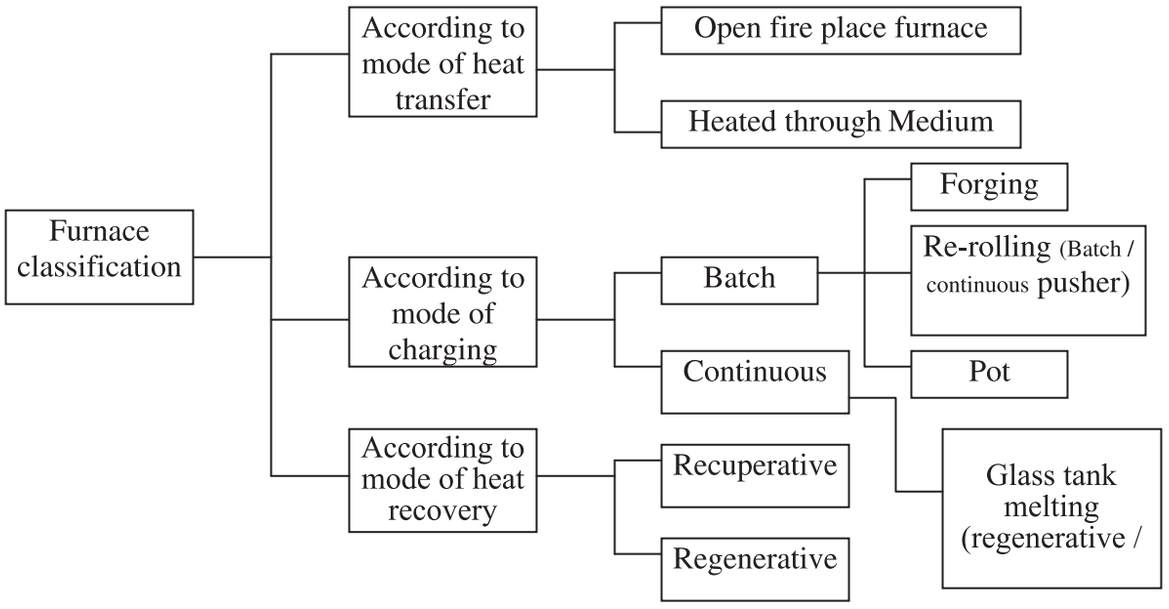

Figure 4.1 Furnace Classification

The electric furnaces can be broadly classified as resistance type for heating and induction and arc for melting of metals.

Characteristics of an Efficient Furnace

Furnace should be designed so that in a given time, as much of material as possible can be heated to a uniform temperature as possible with the least possible fuel and labour. To achieve this end, the following parameters can be considered.

- Determination of the quantity of heat to be imparted to the material or charge.
- Liberation of sufficient heat within the furnace to heat the stock and overcome all heat losses.
- Transfer of available part of that heat from the furnace gases to the surface of the heating stock.
- Equalisation of the temperature within the stock.
- Reduction of heat losses from the furnace to the minimum possible extent.

Furnace Energy Supply

Since the products of flue gases directly contact the stock, type of fuel chosen is of importance. For example, some materials will not tolerate sulphur in the fuel. Also use of solid fuels will generate particulate matter, which will interfere with the stock place inside the furnace. Hence, vast majority of the furnaces use liquid fuel, gaseous fuel or electricity as energy input. Electricity is used in induction and arc furnaces for melting steel and cast iron. Non-ferrous melting utilizes oil as fuel.

Oil Fired Furnace

Furnace oil is the major fuel used in oil fired furnaces, especially for reheating and heat treatment of materials. LDO is used in furnaces where presence of sulphur is undesirable. The key to efficient furnace operation lies in complete combustion of fuel with minimum excess air.

Furnaces operate with efficiencies as low as 7% as against upto 90% achievable in other combustion equipment such as boiler. This is because of the high temperature at which the furnaces have to operate to meet the required demand. For example, a furnace heating the stock to 1200 ^{o}C will have its exhaust gases leaving at least at 1200 ^{o}C resulting in a huge heat loss through the stack. However, improvements in efficiencies have been brought about by methods such as preheating of stock, preheating of combustion air and other waste heat recovery systems.

Typical Furnace System

i) Forging Furnaces

The forging furnace is used for preheating billets and blooms to attain a 'forge' temperature. The furnace temperature is maintained at around 1200 to 1250 ^{o}C. Forging furnaces use an open fireplace system and most of the heat is transmitted by radiation.

The total operating cycle can be divided into (a) heat up time (b) soaking time and (c) forging time. Normally, large pieces are soaked for 4 to 6 hrs inside the furnace to attain uniform temperature

throughout the cross-section of the material. The actual soaking time varies with the type and thickness of the material. The completely soaked material is withdrawn from furnace to the hammer to be forged as required. Larger pieces may have to be reheated several times. A typical forging furnaces is illustrated in Figure 4.2.

Figure 4.2 Forging Furnace

The charging and discharging of the material is done manually and this results in significant heat loss during the forging operation. Forging furnaces use an open fireplace system and most of the heat is transmitted by radiation. Specific fuel consumption depends upon the type of material and number of 'reheats' required.

ii) Rerolling Mill Furnace

a) Batch type

A box type furnace is employed for batch type rerolling mill. The furnace is basically used for heating up scrap, small ingots and billets weighing 2 to 20 kg for rerolling. The charging and discharging of the 'material' is done manually and the final product is in the form of rods, strips etc. The operating temperature is about 1200 ^{o}C. The total cycle time can be further categorized into heat-up time and rerolling time. During heat-up time the material gets heated upto the required temperature and is removed manually for rerolling. The average output from these furnaces varies from 10 to 15 tonnes / day.

b) Continuous Pusher Type

The process flow and operating cycles of a continuous pusher type is the same as that of the batch furnace. The operating temperature is about 1250 ^{o}C. Generally, these furnaces operate 8 to 10 hours

with an output of 50 to 100 tonnes per day. The material or stock recovers a part of the heat in flue gases as it moves down the length of the furnace. Heat absorption by the material in the furnace is slow, steady and uniform throughout the cross-section compared with batch type.

iii) Continuous Steel Reheating Furnaces

The main function of a reheating furnace is to raise the temperature of a piece of steel, typically to between 900 °C and 1250 °C, until it is plastic enough to be pressed or rolled to the desired section, size or shape, The furnace must also meet specific requirements and objectives in terms of stock heating rates for metallurgical and productivity reasons. In continuous reheating, the steel stock forms a continuous flow of material and is heated to the desired temperature as it travels through the furnace.

All furnaces possess the features shown in Figure 4.3

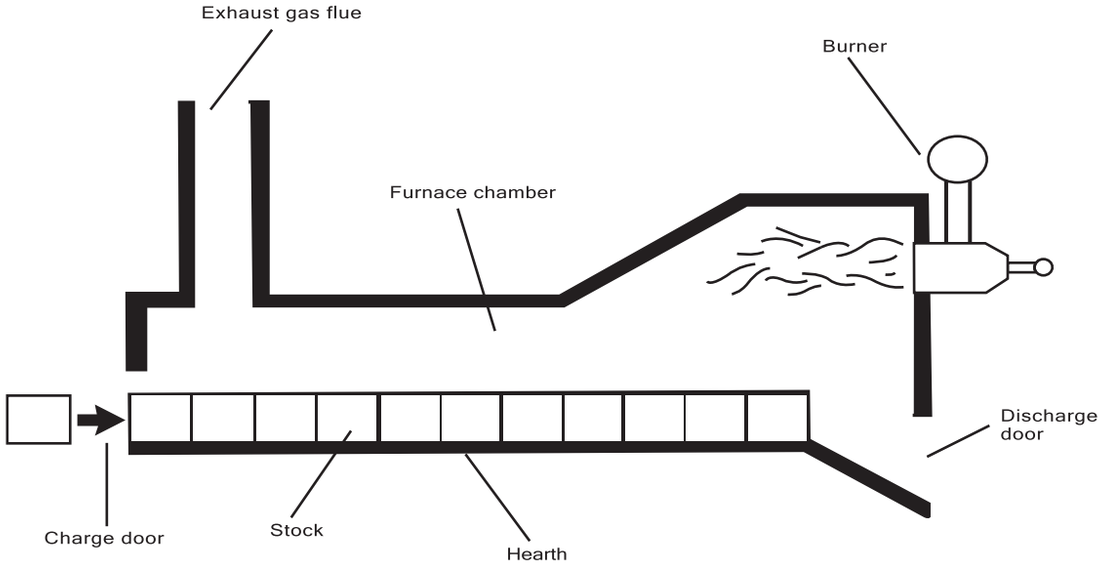

Figure 4.3 Furnace Feature

- A refractory chamber constructed of insulating materials for retaining heat at the high operating temperatures.
- A hearth to support or carry the steel. This can consist of refractory materials or an arrangement of metallic supports that may be water-cooled.
- Burners that use liquid or gaseous fuels to raise and maintain the temperature in the chamber. Coal or electricity can be used for reheating. A method of removing the combustion exhaust gases from the chamber
- A method of introducing and removing the steel from the chamber.
- These facilities depend on the size and type of furnace, the shape and size of the steel being processed, and the general layout of the rolling mill.
- Common systems include roller tables, conveyors, charging machines and furnace pushers.

Heat Transfer in Furnaces

The main ways in which heat is transferred to the steel in a reheating furnace are shown in Figure 4.4**.** In simple terms, heat is transferred to the stock by:

- ✓ Radiation from the flame, hot combustion products and the furnace walls and roof;
- ✓ Convection due to the movement of hot gases over the stock surface.

At the high temperatures employed in reheating furnaces, the dominant mode of heat transfer is wall radiation. Heat transfer by gas radiation is dependent on the gas composition (mainly the carbon dioxide and water vapour concentrations), the temperature and the geometry of the furnace.

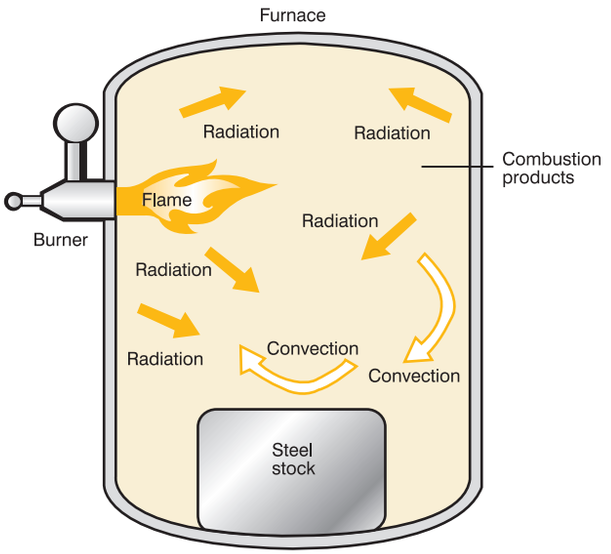

Figure 4.4 Heat Transfer in furnace

Types of Continuous Reheating Furnace

Continuous reheating furnaces are primarily categorised by the method by which stock is transported through the furnace. There are two basic methods:

- Stock is butted together to form a stream of material that is pushed through the furnace. Such furnaces are called pusher type furnaces.
- Stock is placed on a moving hearth or supporting structure which transports the steel through the furnace. Such types include walking beam, walking hearth, rotary hearth and continuous recirculating bogie furnaces.

The major consideration with respect to furnace energy use is that the inlet and outlet apertures should be minimal in size and designed to avoid air infiltration.

i) Pusher Type Furnaces

The pusher type furnace is popular in steel industry. It has relatively low installation and maintenance costs compared to moving hearth furnaces. The furnace may have a solid hearth, but it is also possible to push the stock along skids with water-cooled supports that allow both the top and bottom faces of the stock to he heated. The design of a typical pusher furnace design is shown schematically in Figure 4.5.

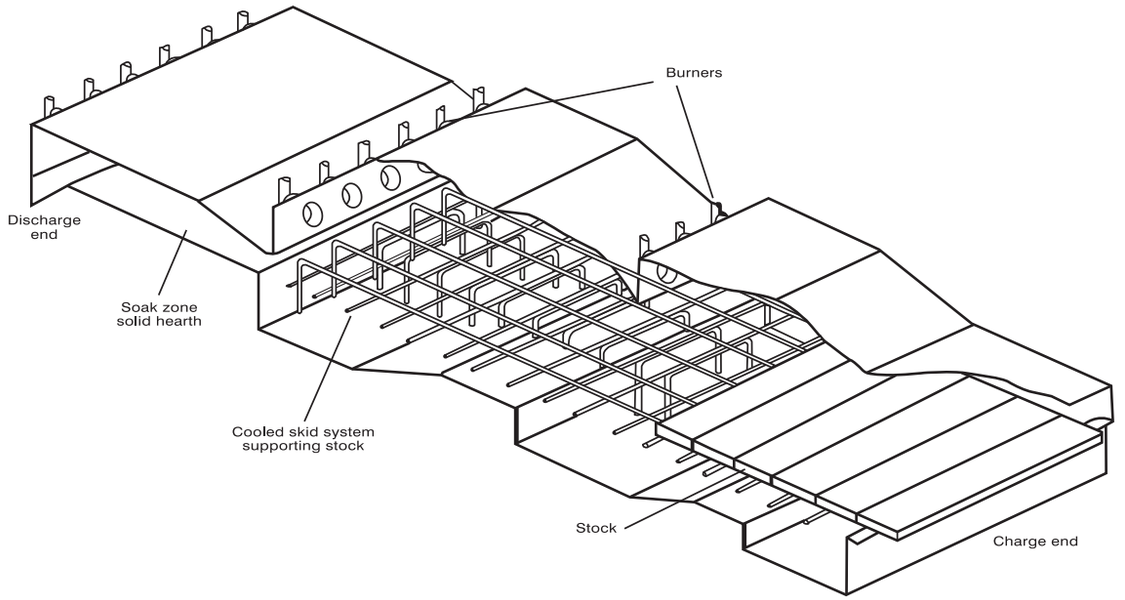

Figure 4.5: Pusher Type Furnaces

Pusher type furnaces, however, do have some disadvantages, including:

- ✓ Frequent damage of refractory hearth and skid marks on material
- ✓ Water cooling energy losses from the skids and stock supporting structure in top and bottom fired furnaces have a detrimental effect on energy use;
- ✓ Discharge must be accompanied by charge:
- ✓ Stock sizes and weights and furnace length are limited by friction and the possibility of stock pile-ups.
- ✓ All round heating of the stock is not possible.

ii) Walking Hearth Furnaces

The walking hearth furnace (Figure.4.6) allows the stock to be transported through the furnace in discrete steps. Such furnaces have several attractive features, including: simplicity of design, ease of construction, ability to cater for different stock sizes (within limits), negligible water cooling energy losses and minimal physical marking of the stock.

The main disadvantage of walking hearth furnaces is that the bottom face of the stock cannot be heated. This can he alleviated to some extent by maintaining large spaces between pieces of stock. Small spaces between the individual stock pieces limits the heating of the side faces and increases the potential for unacceptable temperature differences within the stock at discharge. Consequently, the stock residence time may be long, possibly several hours; this may have an adverse effect on furnace flexibility and the yield may be affected by scaling.

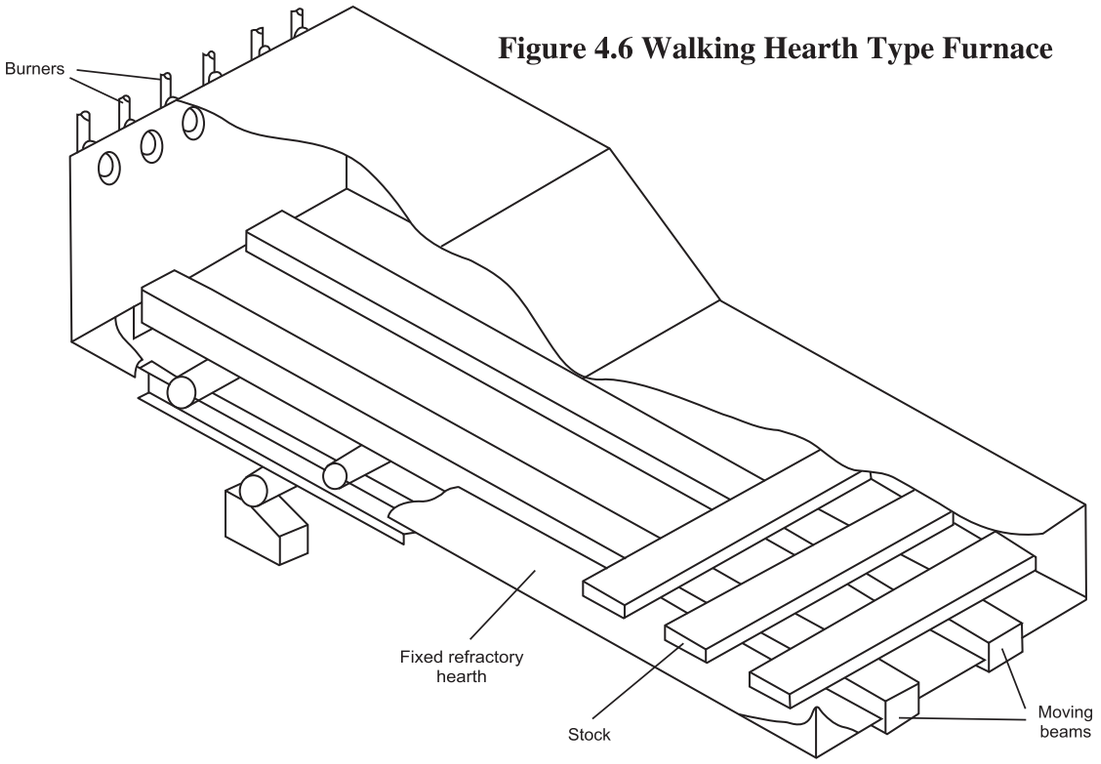

Figure 4.6 Walking Hearth Type Furnace

iii) Rotary hearth furnace

The rotary hearth furnace (Figure 4.7) has tended to supersede the re-circulating bogie type. The heating and cooling effects introduced by the bogies are eliminated, so heat storage losses are less. The rotary hearth has, however a more complex design with an annular shape and revolving hearth.

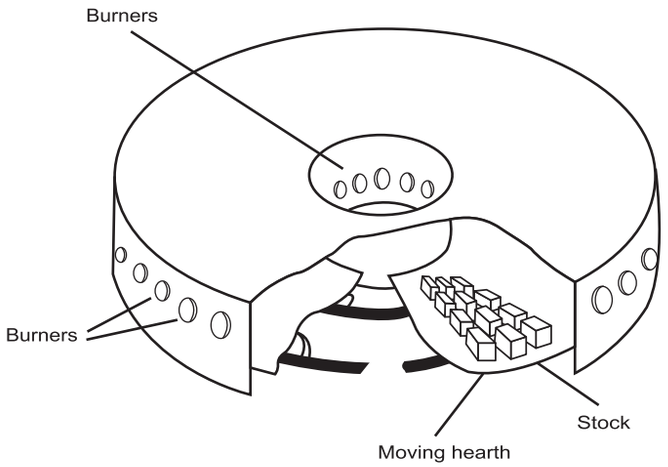

Figure 4.7 Rotary Hearth Furnace

iv) Continuous Recirculating Bogie type Furnaces

These types of moving hearth type furnaces tend to be used for compact stock of variable size and geometry. In bogie furnaces (Figure 4.8), the stock is placed on a bogie with a refractory hearth, which travels through the furnace with others in the form of a train. The entire furnace length is always occupied by bogies. Bogie furnaces tend to be long and narrow and to suffer from problems arising from inadequate sealing of the gap between the bogies and furnace shell, difficulties in removing scale, and difficulties in firing across a narrow hearth width.

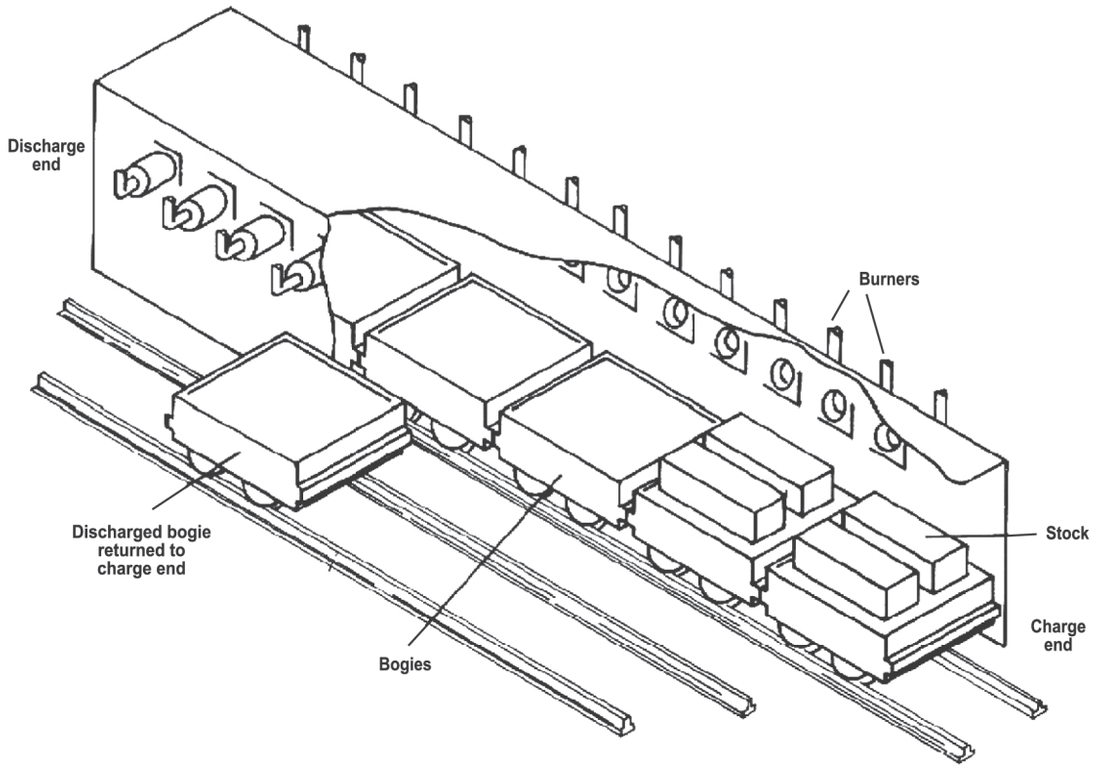

Figure 4.8 Continuous Circulating Bogie Type Furnace

v) Walking Beam Furnaces: The walking beam furnace (Figure 4.9) overcomes many of the problems of pusher furnaces and permits heating of the bottom face of the stock. This allows shorter stock heating times and furnace lengths and thus better control of heating rates, uniform stock discharge temperatures and operational flexibility. In common with top and bottom fired pusher furnaces, however, much of the furnace is below the level of the mill; this may be a constraint in some applications.

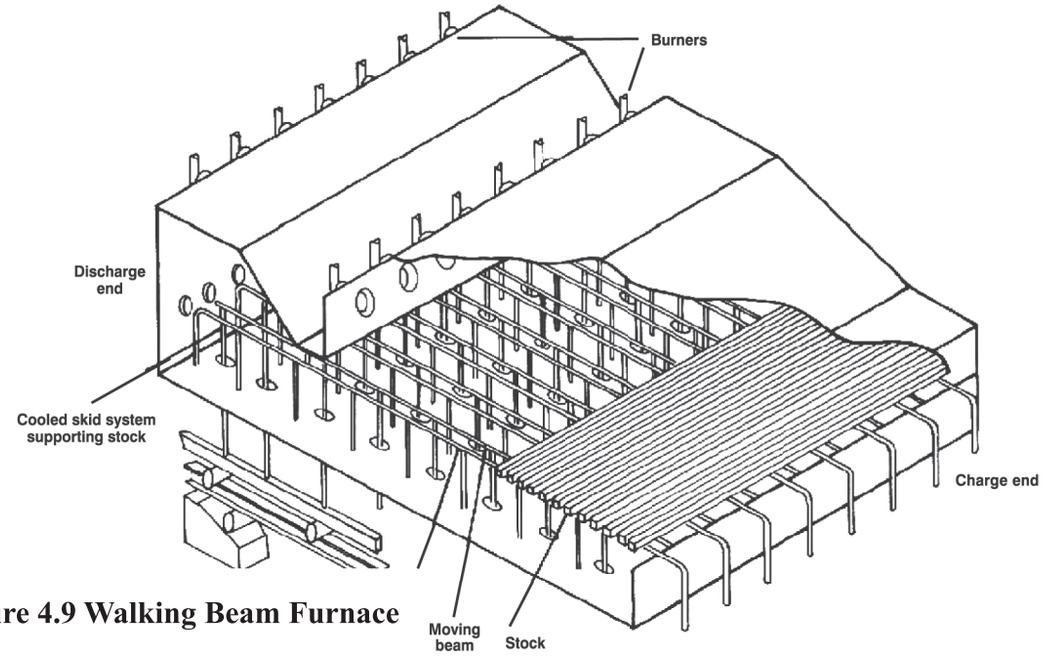

Figure 4.9 Walking Beam Furnace

vi) Cupola Furnace

A cupola furnace (Figure 4.10) is tall, cylindrical melting device used in foundries to melt pig iron, cast iron, foundry returns. The charge used in cupola furnace consists of alternate layers of coke, flux and metal (iron). These three components are continuously built into the cupola furnace. The most commonly used iron - to - coke ratio is 8:1. The flux may be limestone ($CaCO_3$), fluorspar, sodium carbonate or calcium carbide. Limestone is the commonly employed flux. The total weight of the flux will be approximately 1/5th the weight of the coke charge. Sufficient air is passed through the tuyeres for proper combustion of coke.

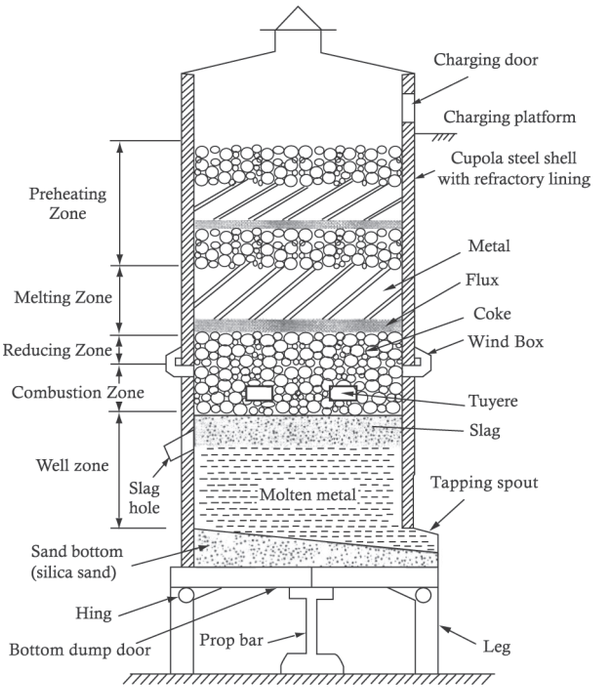

Figure 4.10 Cupola Furnace

Melting in Cupola Furnace

Cupola furnace works on the counter current principle. As the combustion takes place, the charge materials (coke, flux and metal) will be descending downwards, while the hot gases due to combustion will be ascending upwards. Heat exchange takes place between the rising hot gases and the descending charge thereby melting the metal. The liquid metal drops down, while the coke floats up on top of it.

The flux also melts and reacts with the impurities of the molten metal forming a slag. The slag floats on the surface of the molten metal thereby preventing oxidation of the metal.

Zones of Cupola Furnace

Well zone in Cupola Furnace:-

Well zone in the portion situated between the rammed sand bottom and just below the bottom edge of the tuyeres. The molten metal is occupied in this zone.

Combustion Zone in Cupola Furnace

The combustion zone or oxidizing zone is situated normally 15-30 cm from the bottom edge of the tuyeres. It is in this zone where rapid combustion of coke takes place due to which a lot of heat is generated in the furnace. The combustion is rapid due to the supply of blast air through the tuyeres.

Oxidation of manganese and silicon evolve still more heat. The reactions which take place in this zone are:

$C + O_2$ (from air) → CO_2 + heat

$2\ Mn + O_2$ → $2\ MnO$ + heat

$Si + O_2$ → SiO_2 + heat

The temperature in this zone varies from 155 ^{0}C – 185 ^{0}C

Reducing zone or protection zone is the portion located from the top of the combustion zone to the top of the coke bed. In this zone, some of the hot CO_2 gas moving upward through the hot coke gets reduced to CO. In other words, reduction of CO_2 to CO occurs in this zone. Due to the reducing atmosphere, the charge is protected from oxidation. The reaction taking place in this zone is given by.

CO_2 + C (coke) → 2 CO – Heat

Due to the reduction, the temperature reduces to around 1200 ^{o}C in this zone.

Melting Zone in Cupola Furnace

The portion located just above the coke bed to the top of the metal (iron) is called the melting zone. The metal starts melting in this zone and trickles down through the coke bed to the well zone. The molten iron while passing down through the reducing zone picks up carbon and the reaction is given by:

$3\ Fe + 2\ CO$ → $Fe_3C + CO_2$

Preheating zone in Cupola Furnace

The portion occupied from the top surface of the melting zone to the charging door is called 'Preheating zone'. The hot gases rising upwards from the combustion and reducing zone gives its heat to the charge before passing out of the furnace. Thus, the charge is preheated before descending downwards

Hot Blast Cupola

The temperature of exhaust gas of a cupola is as high as 800 °C, making it possible to preheat blast air up to as high as 400 °C by heat exchange. In addition, both the sensible and latent heat of exhaust gas can be recycled for preheating blast air by combustion of CO gas included in exhaust gas. When blast air is preheated to 300 °C or higher, the sensible heat of blast air is added to heat input, activating combustion of coke, leading to the rise in combustion temperature.

vii) Induction Furnaces

Induction furnaces are ideal for melting and alloying a wide variety of metals with minimum melt

losses, however, little refining of the metal is possible. There are two main types of induction furnace: coreless and channel, the principle of operation of which are the same.

Coreless Induction Furnace

Coreless induction furnace (Figure 4.11) consists of: a water cooled helical coil made of a copper tube, a crucible installed within the coil and supporting shell equipped with trunnions on which the furnace may tilt. Alternating current passing through the coil induces alternating currents in the metal charge loaded to the crucible. These induced currents heat the charge.

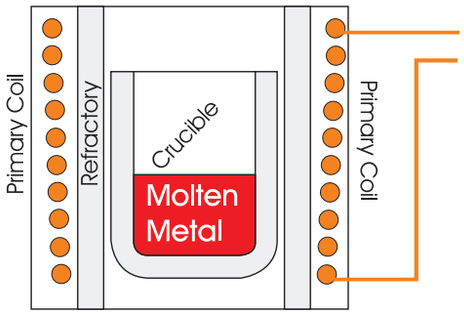

Figure 4.11 Induction Furnace

When the charge is molten, electromagnetic field produced by the coil interacts with the electromagnetic field produced by the induced current. The resulted force causes stirring effect helping homogenizing the melt composition and the temperature.

The frequency of the alternating current used in induction furnaces may vary from the line frequency (50 Hz or 60 Hz) to high frequency 10,000 Hz

The total absolute energy required to melt one tonne of different metals at different molten temperature is given in table 4.1.

Table 4.1. Energy Required to Melt One Tonne of Different Metals

Type of metal	Specific heat require	Latent heat require	Total require kWh/Ton
Mild steel @ 1650 ºC Melting temp.	1000 x 0.682 x 1620 ^{0}C ÷ 3600 kWh ΔT = 1650^{0}C-30^{0}C Specific heat = 0.9 kJ/kg ^{0}C = 307 kWh	272 x 1000 ÷ 3600 kWh Latent Heat = 272 kJ/kg = 76 kWh	307 kWh + 76 kWh = 376 kWh
Aluminum @ 710 ºC Melting temp.	1000 x 0.9 x 680 ^{0}C ÷ 3600 kWh ΔT= 710 ^{0}C-30 ^{0}C Specific heat = 0.9 kJ/kg ^{0}C = 170 kWh	396.9 x 1000 ÷ 3600 kWh Latent Heat = 396.9 kJ/kg = 110 kWh	170 kWh + 110 kWh = 180 kWh
Copper @ 1130 ºC Melting temp.	1000 x 0.386 x 1100 ^{0}C ÷ 3600 kWh ΔT=1130 ^{0}C-30 ^{0}C Specific heat = 0.386 kJ/kg ^{0}C = 118 kWh	212 x 1000 ÷ 3600 kWh Latent Heat = 212 kJ/kg = 59 kWh	118 kWh + 59 kWh = 117 kWh

Gold @ 1130 ºC Melting temp.	1000 x 0.131 x 1130^{0}C ÷ 3600 kWh ΔT=1130 ^{0}C-30 ^{0}C Specific heat = 0.131 kJ/kg ^{0}C = 36.38 kWh	67.62 x 1000 ÷ 3600kWh Latent Heat = 67.62 kJ/kg = 18.78 kWh	36.38 kWh + 18.78 kWh = 56 kWh

Furnace Efficiency

$$Efficiency, \% = \frac{Theoretical\ total\ heat\ required\ for\ melting\ (H_T),\ kWh}{Actual\ Electricity\ consumed\ for\ melting\ (H_A),\ kWh} \times 100$$

Theoretical heat required for melting (H_T)

$$Heat\ required\ for\ melting\ metal, H_1, kWh = \frac{W_m \times \left(C_p \times (T_2 - T_1) + h\right)}{3600}$$

(3600 kJ = 1 kWh)

$$Heat\ required\ for\ melting\ slag, H_2, kWh = \frac{1.65 \times W_s}{3.6}$$

(3.6 MJ = 1 kWh)

Where,

$\mathbf{W_m}$ - Weight of the metal, kg
$\mathbf{W_s}$ - Weight of the slag, kg
C_p - Specific heat of metal, kJ/kg ^{o}C
$\mathbf{T_2}$ - **Final temperature of the metal**
$\mathbf{T_1}$ - Initial or charge temperature of the metal

Total theoretical heat required for melting, $H_T = H_1 + H_2$

Actual electricity consumed for melting (H_A)

The actual consumption of electricity for melting can be measured from the input busbar to the furnace. the difference between Actual and theoretical values will be loss due to conduction, radiation and other losses.

Example – Melting of M.S. scrap

Calculate the furnace efficiency from the data given below

Specific heat - 0.682 kJ/kg ^{0}C
Latent heat - 272 kJ/kg
Melting temperature - 1650 ^{0}C

Charge temperature - 30 ^{0}C
Quantity of metal - 1000 kg
Quantity of slag - 25 kg
Electricity consumed - 625 kWh

Solution

$$Heat\ required\ for\ melting\ metal, H_1, kWh = \frac{1000 \times (0.682 \times (1650 - 30) + 272)}{3600}$$
$$= 382.45 \text{ kWh}$$

$$Heat\ required\ for\ melting\ slag, H_2, kWh = \frac{1.65 \times 25}{3.6}$$
$$= 11.45 \text{ kWh}$$

Total theoretical heat required for melting, H_T = 382.45 + 11.45
= 394 kWh

$$Efficiency, \% = \frac{394,\ kWh}{625,\ kWh} \times 100$$
$$= 63\ \%$$

Distribution of losses in induction furnace

Losses in induction furnace

The theoretical energy require to melt one Ton of steel is 385 TO 400 kWh/Ton. However in actual practice, the specific energy consumption is remarkably higher to 550 - 950 kWh/ton.

1. Power loss in generator / panel = 2 - 4 %
2. Power loss in capacitor Bank = 1.0 - 3 %
3. Power loss in Crucible = 18 - 25 %
   (Water cooled cables, Bus bar, and change over switches)
4. Radiation loss = 7 - 9%

Factors affecting the furnace efficiency

The factors, which affect the furnace efficiency, are as under;

- Due to poor maintenance the total production stops sometimes. Higher breakdown results in increasing the' cost of production
- Due to low supply voltage the furnace draws less power, causes slow melting and inefficient operation resulting increase in production cost.
- Some time lining material selection is wrong with respect to metal. Basic lining is better conductor of heat compared to acidic lining. Wrong lining selection increases breakdown, Furnace down time and furnace losses, resulting in inefficient operation.

- Poor coordination between melting staff & contractor
- Absence of material handling equipment
- Poor molding efficiency, so furnace on hold
- Poor quality scrap, reducing lining life, takes more time to melt.
- Absence of thermal insulation between lining & coil.

Hot Air Generator

Hot air at a wide range of temperatures and pressures is produced for applications like foundry sand drying, shell sand coating, core drying rooms or ovens, drying and processing of ores and minerals, food processing, tea drying, seed drying and paint drying. Furnace oil, HSD, LDO, LPG or natural gas may be used.

Hot air generators supply air heated to elevated temperature by mixing it with products of combustion from a burner. The hot air generator consists of an inner refractory chamber, venturi section where hot products of combustion mix with dilution air, and outlet section.

The combustion chamber is lined with firebrick, while the mixing chamber and the outlet section are lined with insulation brick. Air and oil pipelines required for combustion are included. The combustion equipment comprises of burner with burner block, mounting plate, isolating valve, and oil pumping and heating unit.

4.2 Performance Evaluation of a Typical Fuel Fired Furnace

Thermal efficiency of process heating equipment, such as furnaces, ovens, heaters, and kilns is the ratio of heat delivered to a material and heat supplied to the heating equipment.

In the context of performance assessment of furnaces, the concepts of excess air and computation of various heat losses, as in case of boilers, could be applied, but there exist several additionalities and features, that differentiate furnace efficiency from boiler efficiency. Whereas boiler efficiency of 70-90% is achievable, furnace efficiencies at times are as low as 5%, while with favorable condition, like good design, loading**,** Waste heat recovery O&M practices, efficiencies up to 60% & higher can be achieved. The reasons for low furnace efficiency include;

- High stack temperature and excess air levels.
- Low capacity utilization/ hearth loading.
- Radiation losses due to opening.
- Surface heat losses.
- Batch operation involving heating, cooling and soaking cycles.

One reason for the difference in thermal efficiency between boiler furnaces and industrial furnaces lies in the final temperature of the material being heated. Gases can give up heat to the charge only as long as they are hotter than the charge. In consequence, the flue gases must leave industrial furnace at a very high temperature except for a short time after a cold start.

The purpose of a heating process is to introduce a certain amount of thermal energy into a product, raising it to a certain temperature to prepare it for additional processing or change its properties. To carry this out, the product is heated in a furnace. This results in energy losses in different areas and forms as shown in sankey diagram Figure 4.12. For most heating equipment, a large amount of the heat supplied is wasted in the form of exhaust gases.

These furnace losses include:

- Heat storage in the furnace structure
- Losses from the furnace outside walls or structure
- Heat transported out of the furnace by the load conveyors, fixtures, trays, etc.
- Radiation losses from openings, hot exposed parts, etc.
- Heat carried by the cold air infiltration into the furnace
- Heat carried by the excess air used in the burners.

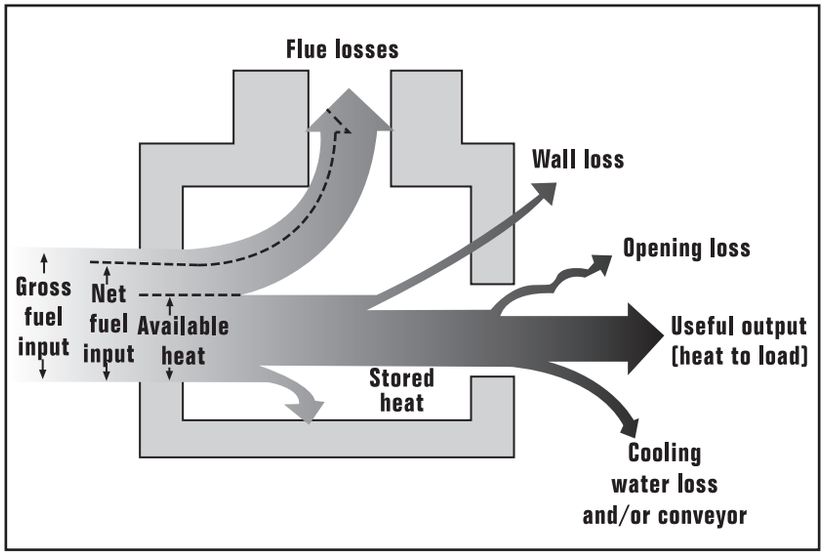

Figure 4.12 Heat losses in industrial heating Furnaces

Stored heat loss: First, the metal structure and insulation of the furnace must be heated so their interior surfaces are about the same temperature as the product they contain. This stored heat is held in the structure until the furnace shuts down, then it leaks out into the surrounding area. The more frequently the furnace is cycled from cold to hot and back to cold again, the more frequently this stored heat must be replaced. Fuel is consumed with no useful output.

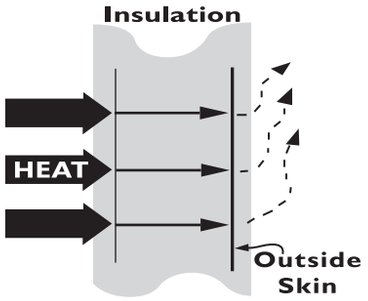

Figure 4.13 Wall Losses

Wall losses: Additional heat losses take place while the furnace is in production. Wall or transmission losses are caused by the conduction of heat through the walls, roof, and floor of the heating device, as shown in Figure 4.13. Once that heat reaches the outer skin of the furnace and radiates to the surrounding area or is carried away by air currents, it must be replaced by an equal amount taken from the combustion gases. This process continues as long as the furnace is at an elevated temperature.

Material handling losses: Many furnaces use equipment to convey the work into and out of the heating chamber, and this can also lead to heat losses. Conveyor belts or product hangers that enter the heating chamber cold and leave it at higher temperatures drain energy from the combustion gases. In car bottom furnaces, the hot car structure gives off heat to the room each time it rolls out of the furnace to load or remove work. This lost energy must be replaced when the car is returned to the furnace.

Cooling media losses. Water or air cooling protects rolls, bearings, and doors in hot furnace environments, but at the cost of lost energy. These components and their cooling media (water, air, etc.) become the conduit for additional heat losses from the furnace. Maintaining an adequate flow of cooling media is essential, but it might be possible to insulate the furnace and load from some of these losses.

Radiation (opening) losses. Furnaces and ovens operating at temperatures above 540°C might have significant radiation losses, as shown in Figure 4.14 Hot surfaces radiate energy to nearby colder surfaces, and the rate of heat transfer increases with the fourth power of the surface's absolute temperature. Anywhere or anytime there is an opening in the furnace enclosure, heat is lost by radiation, often at a rapid rate.

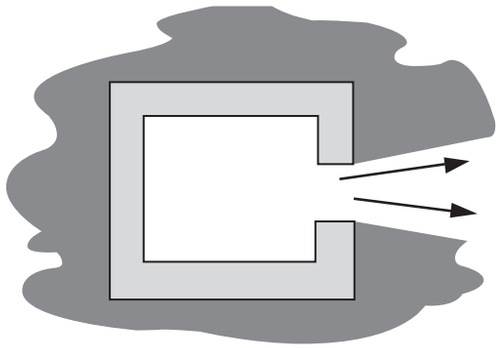

Figure 4.14 Radiation Loss

Waste-gas losses. Waste-gas loss, also known as flue gas or stack loss, is made up of the heat that cannot be removed from the combustion gases inside the furnace. The reason is heat flows from the higher temperature source to the lower temperature heat receiver.

Air infiltration. Excess air does not necessarily enter the furnace as part of the combustion air supply. It can also infiltrate from the surrounding room if there is a negative pressure in the furnace. Because of the draft effect of hot furnace stacks, negative pressures are fairly common, and cold air slips past leaky door seals, cracks and other openings in the furnace. Figure 4.15 illustrates air infiltration from outside the furnace. Every time the door is opened, considerable amount of heat is lost. Economy in fuel can be achieved if the total heat that can be passed on to the stock is as large as possible.

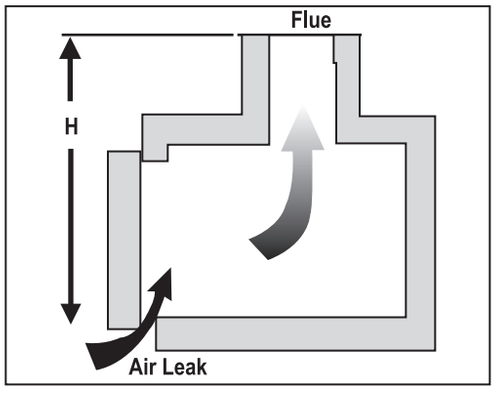

Figure 4.15 Air Infiltration From Furnace Opening.

Direct method

The efficiency of furnace can be assessed by measuring the amount of heat added to the stock and the heat in the fuel consumed, on a batch/day basis as relevant

$$\text{Thermal efficiency of the furnace} = \frac{\text{Heat in the stock}}{\text{Heat in the fuel consumed for heating the stock}} X\ 100$$

The quantity of heat to be imparted (Q) to the stock can be found from the following relation;

$$Q_s = m \times C_p \times (t_1 - t_2)$$

Where

Q_f = Quantity of heat imparted to the stock in kcal
m = Weight of the stock in kg
C_p = Mean specific heat of stock in kcal/kg^oC
t_1 = Final temperature of stock desired, ^oC
t_2 = Initial temperature of the stock before it enters the furnace, ^oC

Heat in fuel = Quantity of fuel (q) X GCV

Example:

The following are the operating parameters of rerolling mill furnace

Weight of input material	-	10 T/hr
Furnace oil consumption	-	600 litres/hr
Specific gravity of oil	-	0.92
Final material temperature	-	1200 ^oC
Initial material temperature	-	40 ^oC
Outlet flue gas temperature	-	650 ^oC
Specific heat of the material	-	0.12 kCal/kg/ ^oC
GCV of oil	-	10,000 kCal/kg
Percentage yield	-	92 %

a. Calculate furnace efficiency by direct method
b. Calculate Specific fuel consumption on finished product basis

Solution:

a) Furnace efficiency by direct method

Heat input	600 lit/hr x 0.92 x 10000 = 55,20,000 kcal/hr
Heat output	10,000 x 0.12 x (1200 – 40) = 1,39,2000 kcal/hr
Efficiency	1,39,2000 /55,20,000 = **25.2 %**

b) Specific fuel consumption on finished product basis

Weight of finished products	10 x 0.92 = 9.2 T/hr
Furnace oil consumption	600 litres/hr
Specific fuel consumption	600/9.2 = **65.2 litres/ton**

Indirect Method

Similar to the method of evaluating boiler efficiency by indirect method (Heat Balance Method), furnace efficiency can also be calculated by indirect methods. Furnace efficiency is calculated after subtracting sensible heat loss in flue gas, loss due to moisture in flue gas, heat loss due to openings in furnace, heat loss through furnace skin and other unaccounted losses.

The detailed calculation by indirect method (Heat balance method) is given in Chapter – 2 of Book – 4. Typical thermal efficiencies for common industrial furnaces are given in Table: 4.2

Table 4.2 Thermal Efficiencies For Common Industrial Furnaces

Furnace Type	Typical thermal efficiencies (%)
1) **Low Temperature furnaces**	
a. 540 – 980 °C (Batch type)	20-30
b. 540 – 980 °C (Continous type)	15-25
c. Coil Anneal (Bell) radiant type	5-7
d. Strip Anneal Muffle	7-12
2) High temperature furnaces	
a. Pusher, Rotary	7-15
b. Batch forge	5-10
3) Continuous Kiln	
a. Hoffman	25-90
b. Tunnel	20-80
4) Ovens	
a. Indirect fired ovens (20°C-370°C)	35-40
b. Direct fired ovens (20°C-370°C)	35-40

The instruments required for carrying out performance evaluation in a furnace is given in the Table 4.3.

Table 4.3 Furnace Instrumentation

Sl. No.	Parameters to be measured	Location of Measurement	Instrument Required	Required Value
1.	Furnace soaking zone temperature (reheating furnaces)	Soaking zone side wall	Pt/Pt-Rh thermocouple with indicator and recorder	1200-1300°C
2.	Flue gas	Flue gas exit from furnace and entry to re-cuperator	Chromel Alummel Thermocouple with indicator	700°C max.
3.	Flue gas	After recuperator	Hg in steel thermometer	300°C (max)
4.	Furnace hearth pressure in the heating zone	Near charging end side wall over hearth level	Low pressure ring gauge	+0.1 mm. of Wg

5.	Flue gas analyser	Near charging end side wall end side	Fuel efficiency monitor for oxygen & temperature.	O_2% = 5 t = 700°C (max)
6.	Billet temperature	Portable	Infrared Pyrometer or optical pyrometer	----

4.3 General Fuel Economy Measures in Furnaces

Typical energy efficiency measures for an industry with furnace are:

1) Complete combustion with minimum excess air
2) Correct heat distribution
3) Operating at the desired temperature
4) Reducing heat losses from furnace openings
5) Maintaining correct amount of furnace draught
6) Optimum capacity utilization
7) Waste heat recovery from the flue gases
8) Minimum refractory losses
9) Use of Ceramic Coatings

1. Complete Combustion with Minimum Excess Air:

The amount of heat lost in the flue gases (stack losses) depends upon amount of excess air. In the case of a furnace carrying away flue gases at 900°C, % heat lost is shown in table 4.4:

Table 4.4 Heat Loss in Flue Gas Based on Excess Air Level

Excess Air	% of total heat in the fuel carried away by waste gases (flue gas temp. 900 °C)
25	48
50	55
75	63
100	71

To obtain complete combustion of fuel with the minimum amount of air, it is necessary to control air infiltration, maintain pressure of combustion air, fuel quality and excess air monitoring.

Higher excess air will reduce flame temperature, furnace temperature and heating rate. On the other hand, if the excess air is less, then unburnt components in flue gases will increase and would be carried away in the flue gases through stack. The figure 4.16 also indicates relation between air ratio and exhaust gas loss.

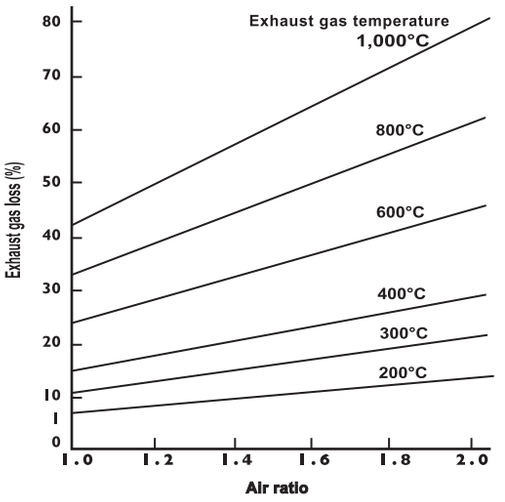

Figure 4.16: Relation Between Air Ratio and Exhaust Gas Loss

The optimization of combustion air is the most

attractive and economical measure for energy conservation. The impact of this measure is higher when the temperature of furnace is high. Air ratio is the value that is given by dividing the actual air amount by the theoretical combustion air amount, and it represents the extent of excess of air.

If a reheating furnace is not equipped with an automatic air/fuel ratio controller, it is necessary to periodically sample gas in the furnace and measure its oxygen contents by a gas analyzer. The Figure 4.17 shows a typical example of a reheating furnace equipped with an automatic air/fuel ratio controller.

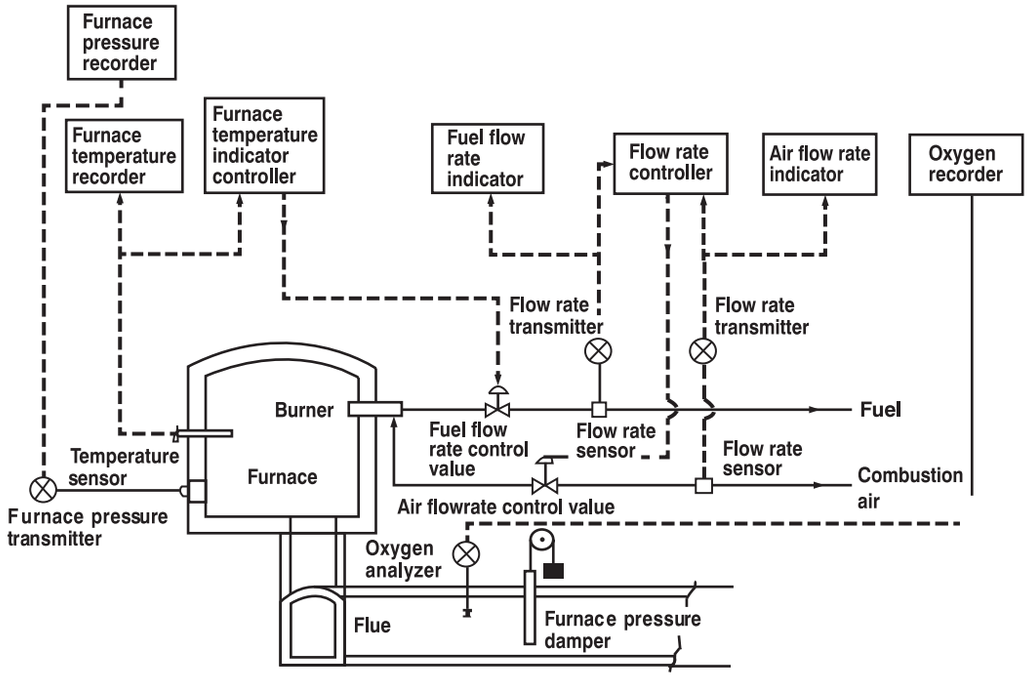

Figure 4.17 Air/Fuel Ratio Control System with Flow Rate Controller

More excess air also means more scale losses, which is equally a big loss in terms of money.

2. Proper Heat Distribution:

Furnace design should be such that in a given time, as much of the stock could be heated uniformly to a desired temperature with minimum fuel firing rate.

Following care should be taken when using burners, for proper heat distribution:

i) The flame should not touch any solid object and should propagate clear of any solid object. Any obstruction will de atomise the fuel particles thus affecting combustion and create black smoke. If flame impinges on the stock, there would be increase in scale losses (Refer Figures 4.18 and 4.19).

ii) If the flames impinge on refractories, the incomplete combustion products can settle and react with the refractory constituents at high flame temperatures.

iii) The flames of different burners in the furnace should stay clear of each other. If they intersect, inefficient combustion would occur. It is desirable to stagger the burners on the opposite sides.

iv) The burner flame has a tendency to travel freely in the combustion space just above the material. In small furnaces, the axis of the burner is never placed parallel to the hearth but always at an upward angle. Flame should not hit the roof.

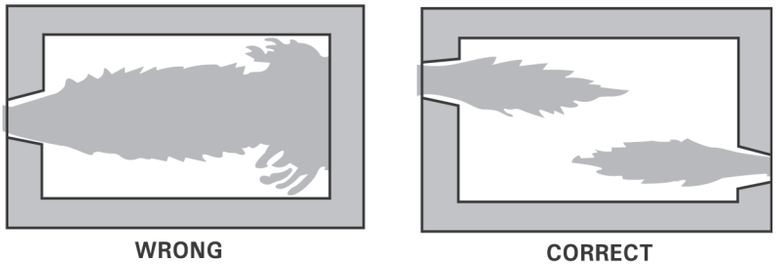

Figure 4.18: Heat Distribution in Furnace

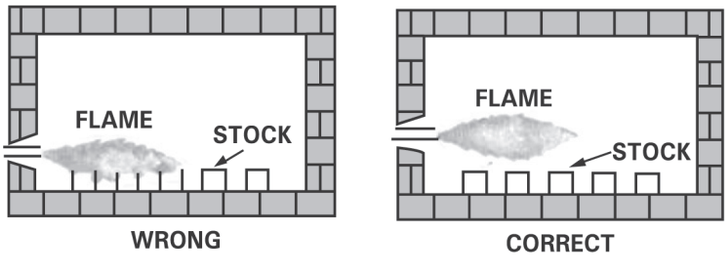

Figure 4.19: Alignment of Burners in Furnace

v) The larger burners produce a long flame, which may be difficult to contain within the furnace walls. More burners of less capacity give better heat distribution in the furnace and also increase furnace life.

vi) For small furnaces, it is desirable to have a long flame with golden yellow colour while firing furnace oil for uniform heating. The flame should not be too long that it enters the chimney or comes out through the furnace top or through doors. In such cases, major portion of additional fuel is carried away from the furnace.

3. Maintaining Optimum Operating Temperature of Furnace :

It is important to operate the furnace at optimum temperature. The operating temperatures of various furnaces are given in table 4.5.

Table 4.5 Operating Temperature of Various Furnaces

Slab Reheating furnaces	1200 °C
Rolling Mill furnaces	1200 °C
Bar furnace for Sheet Mill	800 °C
Bogey type annealing furnaces	650 °C -750 °C

Operating at too high temperatures than optimum causes heat loss, excessive oxidation, de-carbonization as well as over-stressing of the refractories. These controls are normally left to

operator judgment, which is not desirable. To avoid human error, on/off controls should be provided.

4. Prevention of Heat Loss through Openings

Heat loss through openings consists of the heat loss by direct radiation through openings and the heat loss caused by combustion gas that leaks through openings.

The heat loss from an opening can also be calculated using the following formula:

$$Q = 4.88 x \left(\frac{T}{100} \right)^4 x a x A x H$$

where

T: absolute temperature (K)
a: factor for total radiation
A: area of opening, m^2
H: time (Hr)

If the furnace pressure is slightly higher than outside air pressure (as in case of reheating furnace) during its operation, the combustion gas inside may blow off through openings and heat is lost with that. But damage is more, if outside air intrudes into the furnace, making temperature distribution uneven and oxidizing billets. This heat loss is about 1% of the total quantity of heat generated in the furnace, if furnace pressure is controlled properly.

5. Control of furnace draft:

If negative pressures exist in the furnace, air infiltration is liable to occur through the cracks and openings thereby affecting air-fuel ratio control. Tests conducted on apparently airtight furnaces have shown air infiltration up to the extent of 40%. Neglecting furnaces pressure could mean problems of cold metal and non-uniform metal temperatures, which could affect subsequent operations like forging and rolling and result in increased fuel consumption. For optimum fuel consumption, slight positive pressure should be maintained in the furnace as shown in Figure 4.20.

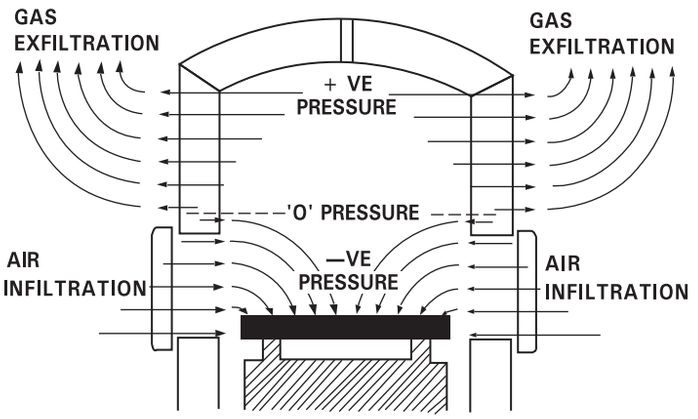

Figure 4.20 Effect of Pressure on the Location of Zero Level and Infiltration of Air

Ex-filtration is less serious than infiltration. Some of the associated problems with ex filtration are leaping out of flames, overheating of the furnace refractories leading to reduced brick life, increased

furnace maintenance, burning out of ducts and equipments attached to the furnace, etc.

In addition to the proper control on furnace pressure, it is important to keep the openings as small as possible and to seal them in order to prevent the release of high temperature gas and intrusion of outside air through openings such as the charging inlet, extracting outlet and peephole on furnace walls or the ceiling.

6. Optimum Capacity Utilization:

One of the most vital factors affecting efficiency is loading. There is a particular loading at which the furnace will operate at maximum thermal efficiency. If the furnace is under loaded a smaller fraction of the available heat in the working chamber will be taken up by the load and therefore efficiency will be low.

The best method of loading is generally obtained by trial-noting the weight of material put in at each charge, the time it takes to reach temperature and the amount of fuel used. Every endeavour should be made to load a furnace at the rate associated with optimum efficiency although it must be realised that limitations to achieving this are sometimes imposed by work availability or other factors beyond control.

The loading of the charge on the furnace hearth should be arranged so that

- It receives the maximum amount of radiation from the hot surfaces of the heating chambers and the flames produced.
- The hot gases are efficiently circulated around the heat receiving surfaces

Stock should not be placed in the following position

- In the direct path of the burners or where flame impingement is likely to occur.
- In an area which is likely to cause a blockage or restriction of the flue system of the furnace.
- Close to any door openings where cold spots are likely to develop.

The other reason for not operating the furnace at optimum loading is the mismatching of furnace dimension with respect to charge and production schedule.

In the interests of economy and work quality the materials comprising the load should only remain in the furnace for the minimum time to obtain the required physical and metallurgical requirements. When the materials attain these properties they should be removed from the furnace to avoid damage and fuel wastage. The higher the working temperature, higher is the loss per unit time. The effect on the materials by excessive residence time will be an increase in surface defects due to oxidation. The rate of oxidation is dependent upon time, temperature, as well as free oxygen content. The possible increase in surface defects can lead to rejection of the product. It is therefore essential that coordination between the furnace operator, production and planning personnel be maintained.

Optimum utilization of furnace can be planned at design stage. Correct furnace for the jobs should be selected considering whether continuous or batch type furnace would be more suitable. For a continuous type furnace, the overall efficiency will increase with heat recuperation from the waste gas stream. If

only batch type furnace is used, careful planning of the loads is important. Furnace should be recharged as soon as possible to enable use of residual furnace heat.

7. Waste Heat Recovery from Furnace Flue Gases:

In any industrial furnace the products of combustion leave the furnace at a temperature higher than the stock temperature. Sensible heat losses in the flue gases, while leaving the chimney, carry 35 to 55 per cent of the heat input to the furnace. The higher the quantum of excess air and flue gas temperature, the higher would be the waste heat availability.

Waste heat recovery should be considered after all other energy conservation measures have been taken. Minimizing the generation of waste heat should be the primary objective.

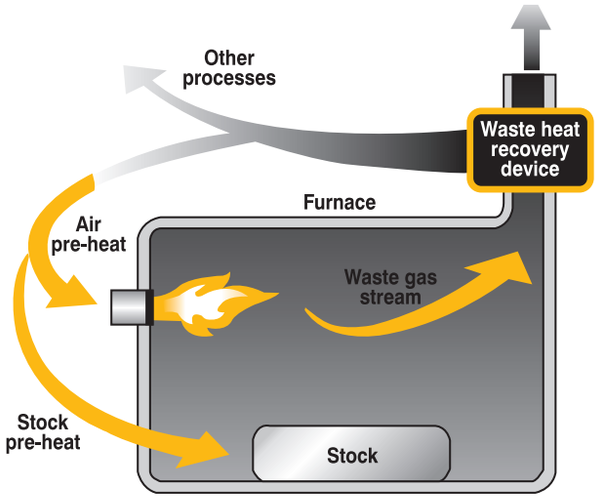

Figure 4.21 Waste Heat Recovery

The sensible heat in flue gases can be generally recovered by the following methods. (Figure 4.21)

- Charge (stock) preheating,
- Preheating of combustion air,
- Utilizing waste heat for other process (to generate steam or hot water by a waste heat boiler)

Charge Pre-heating

When raw materials are preheated by exhaust gases before being placed in a heating furnace, the amount of fuel necessary to heat them in the furnace is reduced. Since raw materials are usually at room temperature, they can be heated sufficiently using high-temperature gas to reduce fuel consumption rate.

Preheating of Combustion Air

For a long time, the preheating of combustion air using heat from exhaust gas was not used except for large boilers, metal-heating furnaces and high-temperature kilns. This method is now being employed in compact boilers and compact industrial furnaces as well. (Refer Figure 4.22)

The energy contained in the exhaust gases can be recycled by using it to pre-heat the combustion air. A variety of equipment is available; external recuperators are common, but other techniques are now available such as self-recuperative burners. For example, with a furnace exhaust gas temperature of

l,000°C, a modern recuperator can pre-heat the combustion air to over 500”C, giving energy savings compared with cold air of up to 30%.

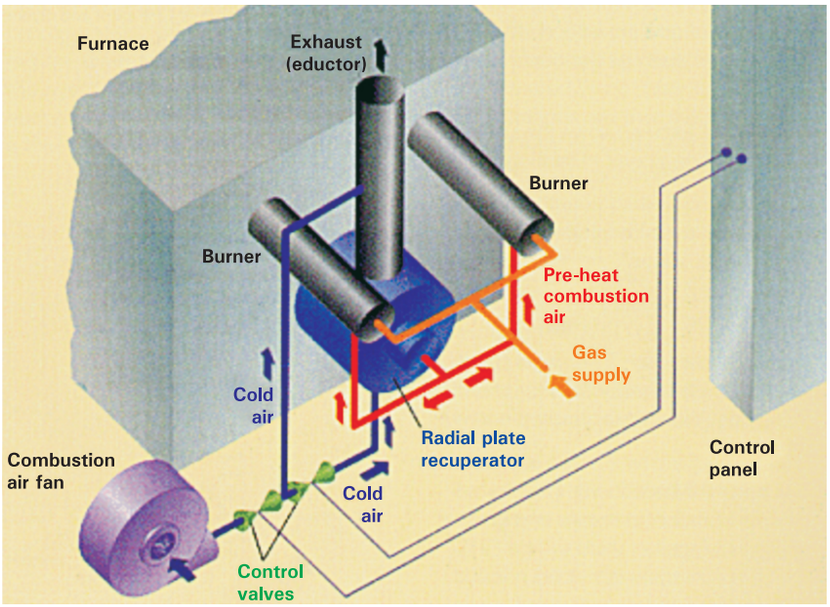

Figure 4.22 Preheating the Air for Combustion by a Recuperator

External *Recuperators*

There are two main types of external recuperators:

- ✓ radiation recuperators;
- ✓ convection recuperators

Radiation recuperators generally take the form of concentric cylinders, in which the combustion air passes through the annulus and the exhaust gases from the furnace pass through the centre, see Figure 23 (a). The simple construction means that such recuperators are suitable for use with dirty gases, have a negligible resistance to flow, and can replace the flue or chimney if space is limited. The annulus can be replaced by a ring of vertical tubes, but this design is more difficult to install and maintain. Radiation recuperators rely on radiation from high temperature exhaust gases and should not he employed with exhaust gases at less than about 800°C.

Convection recuperators consist essentially of bundles of drawn or cast tubes; see Figure 23 (b). Internal and/or external fins can be added to assist heat transfer. The combustion air normally passes through the tubes and the exhaust gases outside the tubes, but there are some applications where this is reversed. For example, with dirty gases, it is easier to keep the tubes clean if the air flows on the outside. Design variations include ‘U’ tube and double pass systems. Convection recuperators are more suitable for exhaust gas temperatures of less than about 900°C.

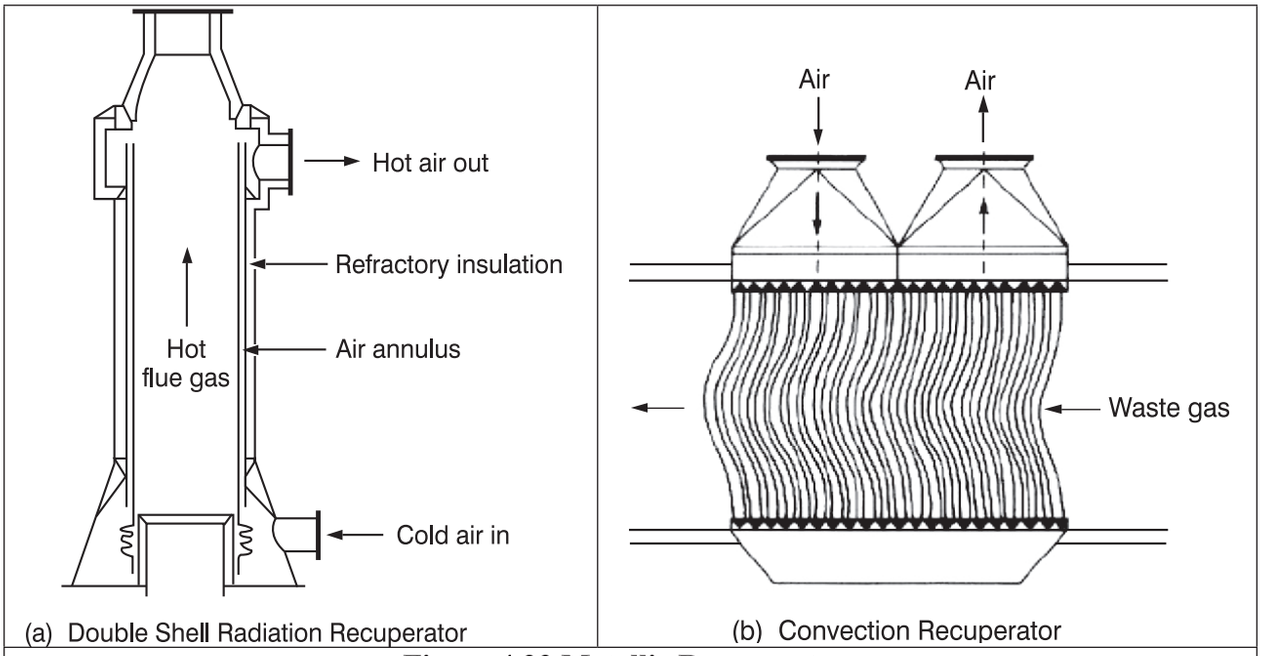

Figure 4.23 Metallic Recuperators

Self Recuperative Burners

Self-recuperative burners (SRBs) are based on traditional heat recovery techniques in that the products of combustion are drawn through a concentric tube recuperator around the burner body and used to pre-heat the combustion air (Figure 4.24.)

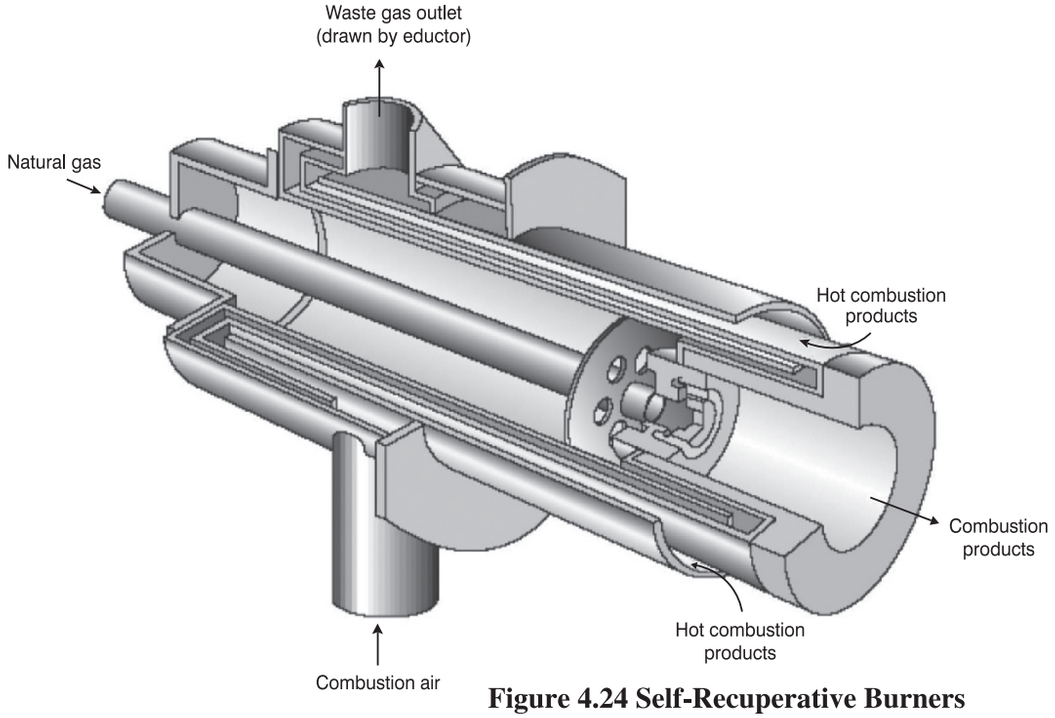

Figure 4.24 Self-Recuperative Burners

A major advantage of this type of system is that it can be retro-fitted to an existing furnace structure to increase production capability without having to alter the existing exhaust gas ducting arrangements. SRBs are generally more suited to heat treatment furnaces where exhaust gas temperatures are lower and there are no stock recuperation facilities.

Estimation of fuel savings

By using preheated air for combustion, fuel can be saved. The fuel saving rate is given by the following formula:

$$S = \frac{P}{F + P - Q} \; x100\,(\%)$$

Where, S: fuel saving rate, %
F: Calorific value of fuel (kcal/kg fuel)
P: quantity of heat brought in by preheated air (kcal/kg fuel)
Q**:** quantity of heat taken away by exhaust gas (kcal/kg fuel)

By this formula, fuel saving rates for heavy oil and natural gas were calculated for various temperatures of exhaust gas and preheated air. The results are shown in the following Figure 4.25 and Figure 4.26.

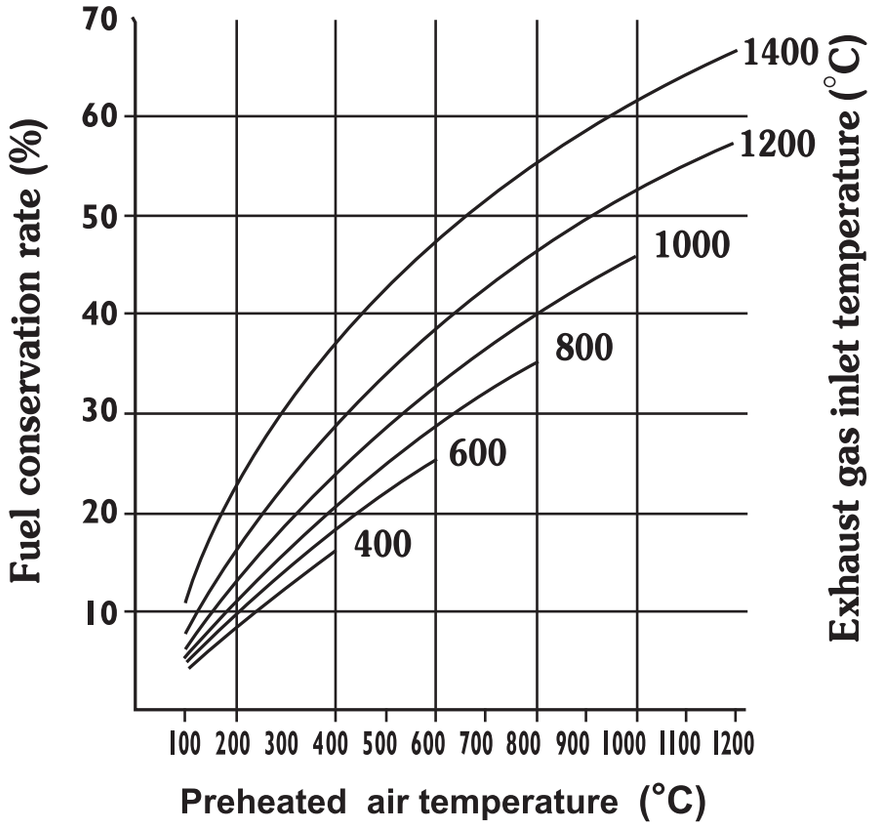

Figure 4.25 Fuel Conservation Rate when Oil is Used

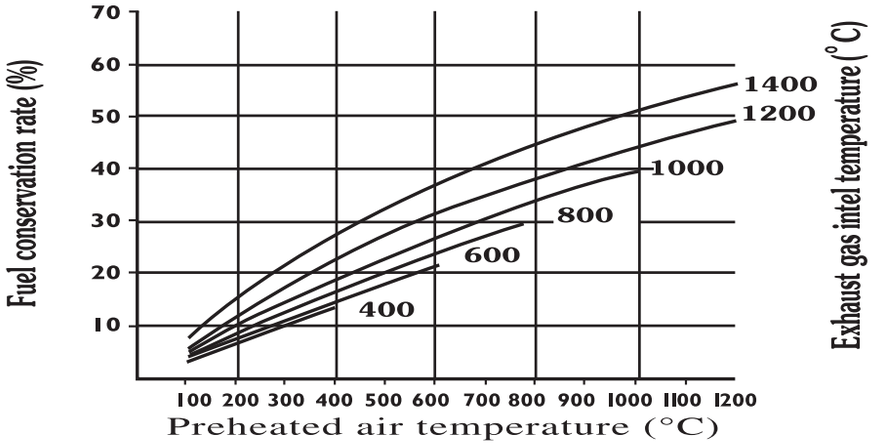

Figure 4.26 Fuel Conservation Rate when Natural Gas is Used

For example, when combustion air for heavy oil is preheated to 400°C by a heat exchanger with an inlet temperature of 800 °C, the fuel conservation rate is estimated to be about 20 percent. When installing a recuperator in a continuous steel reheating furnace, it is important to choose a preheated air temperature that will balance the fuel saving effect and the invested cost for the equipment.

Also, the following points should be checked:

- Draft of exhaust gas: When exhaust gas goes through a recuperator, its draft resistance usually causes a pressure loss of 5-10 mm H_2O. Thus, the draft of stack should be checked.
- Air blower for combustion air: While the air for combustion goes through a recuperator, usually 100-200 mm H_2O pressure is lost. Thus, the discharge pressure of air blower should be checked, and the necessary pressure should be provided by burners.

Since the volume of air is increased owing to its preheating, it is necessary to be careful about the modification of air-duct diameters and blowers. As for the use of combustion gases resulting from high-density oils with a high sulphur content, care must be taken to avoid problems such as clogging with dust or sulphides, corrosion or increases in nitrogen oxides.

Utilizing Waste Heat as a Heat Source for Other Processes

The temperature of heating-furnace exhaust gas can be as high as 400- 600 °C, even after heat has been recovered from it.

When a large amount of steam or hot water is needed in a plant, installing a waste heat boiler to produce the steam or hot water using the exhaust gas heat is preferred. If the exhaust gas heat is suitable for equipment in terms of heat quantity, temperature range, operation time etc., the fuel consumption can be greatly reduced. In one case, exhaust gas from a quenching furnace was used as a heat source in a tempering furnace so as to obviate the need to use fuel for the tempering furnace itself.

8. Minimising Wall Losses:

About 30-40% of the fuel input to the furnace generally goes to make up for heat losses in intermittent or continuous furnaces. The appropriate choice of refractory and insulation materials goes a long way in achieving fairly high fuel savings in industrial furnaces.

The heat losses from furnace walls affect the fuel economy considerably. The extent of wall losses depends on:

- Emissivity of wall
- Thermal conductivity of refractories
- Wall thickness
- Whether furnace is operated continuously or intermittently

Heat losses can be reduced by increasing the wall thickness, or through the application of insulating bricks. Outside wall temperatures and heat losses of a composite wall of a certain thickness of firebrick and insulation brick are much lower, due to lesser conductivity of insulating brick as compared to a refractory brick of similar thickness. In the actual operation in most of the small furnaces the operating periods alternate with the idle periods. During the off period, the heat stored in the refractories during the on period is gradually dissipated, mainly through radiation and convection from the cold face. In addition, some heat is abstracted by air flowing through the furnace. Dissipation of stored heat is a loss, because the lost heat is again imparted to the refractories during the heat "on" period, thus consuming extra fuel to generate that heat. If a furnace is operated 24 hours, every third day, practically all the heat stored in the refractories is lost. But if the furnace is operated 8 hours per day all the heat stored in the refractories is not dissipated. For a furnace with a firebrick wall of 350 mm thickness, it is estimated that 55 percent of the heat stored in the refractories is dissipated from the cold surface during the 16 hours idle period. Furnace walls built of insulating refractories and cased in a shell reduce the flow of heat to the surroundings.

Prevention of Radiation Heat Loss from Surface of Furnace

The quantity of heat release from surface of furnace body is the sum of natural convection and thermal radiation. This quantity can be calculated from surface temperatures of furnace. The temperatures on furnace surface should be measured at as many points as possible, and their average should be used. If the number of measuring points is too small, the error becomes large.

The quantity (Q) of heat release from a reheating furnace is calculated with the following formula:

$$Q = a\,x\,(t_1 - t_2)^{5/4} + 4.88\,E\,x\left(\left(\frac{t_1 + 273}{100}\right)^4 - \left(\frac{t_2 + 273}{100}\right)^4\right)$$

Where, Q: Quantity of heat released (kcal/hr/m^2)

a : Factor regarding direction of the surface of natural convection ceiling = 2.8, side walls = 2.2, hearth = 1.5

t_1 : Temperature of external wall surface of the furnace (°C)

t_2 : Temperature of air around the furnace (°C)

E: Emissivity of external wall surface of the furnace

The first term of the formula above represents the quantity of heat release by natural convection, and the second term represents the quantity of heat release by radiation.

Example

There is a reheating furnace whose ceiling has 20 m^2 of surface area. The average surface temperatures is measured to be 80°C. Evaluate the quantity of heat release from the ceiling.

The total quantity of heat release is

$$Q = a\,x\,(t_1 - t_2)^{5/4} + 4.88\,E\,x\left(\left(\frac{t_1+273}{100}\right)^4 - \left(\frac{t_2+273}{100}\right)^4\right)$$

$$= 2.8\times(80-30)^{5/4} + 4.88\times 0.75\times\left(\left(\frac{80+273}{100}\right)^4 - \left(\frac{30+273}{100}\right)^4\right)$$

= 632 kcal/hr/m^2

Total Heat loss = 632 x 20

= 12640 kcal/hr

Use of Ceramic Fibre

Ceramic fibre is a low thermal mass refractory used in the hot face of the furnace and fastened to the refractory walls. Due to its low thermal mass the storage losses are minimized. This results in faster heating up of furnace and also faster cooling. Energy savings by this application is possible only in intermittent furnaces. More details about ceramic fibre are given in the chapter on insulation and refractories.

9. Use of Ceramic Coatings

Ceramic coatings in furnace chamber promote rapid and efficient transfer of heat, uniform heating and extended life of refractories. The emissivity of conventional refractories decreases with increase in temperature whereas for ceramic coatings it increases. This outstanding property has been exploited for use in hot face insulation.

Ceramic coatings are high emissivity coatings which when applied has a long life at temperatures up to 1350°C. The coatings fall into two general categories-those used for coating metal substrates, and those used for coating refractory substrates. The coatings are non-toxic, non-flammable and water based. Applied at room temperatures, they are sprayed and air dried in less than five minutes. The coatings allow the substrate to maintain its designed metallurgical properties and mechanical strength. Installation is quick and can be completed during shut down. Energy savings of the order of 8-20% have been reported depending on the type of furnace and operating conditions.

10. Fish Bone Diagram for Energy Conservation Analysis in Furnaces

All the possible measures discussed can be incorporated in furnace design and operation. The figure 4.27 shows characteristics diagram of energy conservation for a fuel-fired furnace.

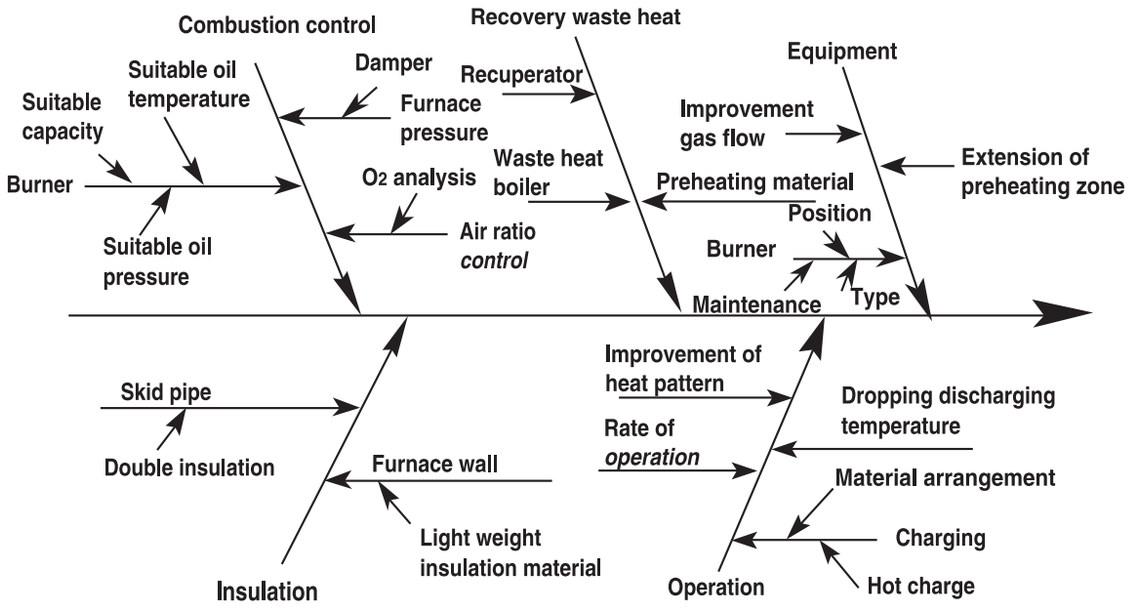

Figure 4.27 Characteristic Diagram of Energy Conservation for Reheating Furnace

Solved Example:

An oil fired reheating furnace has an operating temperature of around 1000°C. Average furnace oil consumption is 440 litres/hour. The flue gas exit temperature after the air preheater is 300°C. Combustion air is preheated from ambient temperature of 35°C to 200°C through the air preheater. The other data are as given below.

Specific gravity of oil = 0.92
Calorific value of oil = 10,200 kcal/kg
Average O_2 percentage in flue gas = 14%
Theoretical air required = 14 kg of air per kg of oil
Specific heat of air = 0.24 kcal/kg°C
Specific heat of flue gas = 0.23 kcal/kg°C

Find out the sensible heat carried away by the exhaust gases and heat recovered by the combustion air in kcal/hr as a percentage of the energy input.

Ans:

Energy input = 440 litres/hr
= 440 x 0.92 kg/hr

= 404.80 kg/hr
= 404.80 x 10,200
= 41,28,960 kcal/hr

Excess air = $\frac{(\%O_2)}{(21-O_{2\%})}$ x 100
= $\frac{14 \times 100}{(21-14)}$
= 200%

Theoretical air required = 14 kg of air to burn 1 kg of oil
Actual mass of air required = 14 x (1 + 200/100) kg/kg of oil
= 42 kg/kg of oil

Sensible heat loss in the flue gas = m x C_p x ΔT
m = mass of flue gas
= 42 + 1
= 43 kg/kg of oil

C_p = Specific heat of flue gas
= 0.23

ΔT = Temperature of flue gas – Ambient Temperature
= 300°C – 35°C
= 265 °C

Heat loss = 43 x 0.23 x (300-35)
= 43 x 0.23 x 265
= 2620.85 kCal/kg of oil
= 2620.85 x 404.80 kcal/hr
= 10,60,920 kcal/hr

Sensible heat loss in the flue gas as % heat loss to input energy
= $\frac{10,60,920 \times 100}{41,28,960}$
= 25.7 %

Heat gained by combustion air = 42 x 0.24 x (200-35)
= 1663.2 kcal/kg of oil
= 1663.2 x 404.80 kcal/hr
= 673263.36 kcal/hr

Heat gained by combustion air as % of input energy
= $\frac{673263.36 \times 100}{41,28,960}$
= 16.3 %

QUESTIONS	
Objective Type Questions	
1.	The emissivity of ceramic coatings used in furnace: a) decreases with increase in temperature b) Increases with increase in temperature c) remains constant d) decreases with increase in furnace pressure
2.	In reheating furnace, scale losses will: a) increase with excess air b) decrease with the excess air c) will have no relation with excess air d) will increase with nitrogen in air
3.	For optimum fuel consumption, the pressure at which furnaces operate should be ___. a) slightly negative b) slightly positive c) neutral d) any of the above
4.	Higher excess air in an oil fired furnace would result in: a) increased furnace temperature b) increased heating rate c) reduced flame temperature d) none of the above
5.	Heat loss through openings in furnaces is directly proportional to a) fourth power of furnace temperature b) square of absolute furnace temperature c) absolute furnace temperature d) fourth power of absolute furnace temperature
6.	Pick up the wrong statement: The thermal efficiency of the furnace increases by a) increasing the furnace loading b) increasing the excess air flow rate c) reducing the surface heat loss d) minimizing the CO loss and unburnt losses
7.	If there is an opening on the furnace body, heat in the furnace escapes to the outside mainly as: a) radiation heat b) conduction heat c) convection heat d) none of the above
8.	Furnace wall heat loss does not depend on a) temperatures of external wall surfaces b) velocity of air around the furnace c) k-value d) stock to be heated.
9.	Which property of ceramic coating influences energy savings in furnaces a) conductivity b) convective heat transfer coefficient c) emissivity d) coating thickness
10.	The efficiency of a reheating furnace, operating at 10 tonnes per hour consuming furnace oil of 230 kg/hour for reheating the material from 40 °C – 1100 °C (consider specific heat of material is 0.13 kcal / kg °C and calorific value of furnace oil is 10,000 kcal /kg) is ___. a) 60 b)70 c)80 d)50

Short Type Questions	
S-1	Define the efficiency of a furnace and explain why furnace efficiencies are lower than boiler efficiencies. State measures to improve the efficiency of furnaces.
S-2	The efficiency of a billet heating furnace with an output of 15 tonne/ hr was 32%. Find out the specific fuel consumption in litres/ tonne of billet heating and total fuel consumption per hour as per data given below: Billet heating furnace: Initial temp. = 50°C Final temp. = 1150°C Specific heat of billet = 0.12 kCal/ kg°C Density of fuel oil = 0.95 kg/ litre GCV of fuel oil = 10,000 kCal/kg Determine the specific fuel consumption in litres/ tonne and total fuel consumption in litres/hr.
S-3	A furnace output is 5000 kg/hour of billets. Thermal efficiency is claimed to be 25% Specific heat of billet is 0.12 kcal/kg °C. Billets enter the furnace at 40°C and leave at 1200°C. Calculate the hourly oil consumption in liter if GCV of oil is 9,200 kcal/liter.
S-4	List any five heat losses occurring in a furnace.
S-5	A reheating furnace output is 10 TPH. The billets are heated from 50°C to 1200°C. The oil consumption rate is 700 liters /hr. Specific gravity of FO is 0.95; Net Calorific value of FO is 9650 kcal/kg, and the specific heat of the billet is 0.12 kcal/kg °C. Find out the thermal efficiency of the furnace on NCV basis.
Long Type Questions	
L-1	(a) Briefly explain why the slight positive draft pressure should be maintained in the furnaces. (b) Explain the significance of achieving optimum capacity utilisation in furnaces.
L-2	In an industry, an electrical furnace consuming 50 kWh/batch is to be converted into furnace oil fired furnace. Estimate the annual savings with furnace oil replacement considering the following: Efficiency of electrical furnace : 75% Efficiency of FO fired furnace (NCV Basis) : 52 % Number of batches : 5000 batches/ year Cost of Electricity : Rs. 4 per kWh Cost of FO : Rs. 20 per kg Net Calorific Value of FO : 9650 kcal /kg

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Chapter 5

Insulation & Refractories

Learning Objectives

In this chapter you will learn about

- Need and benefits of insulation
- Types, materials and applications
- Economic thickness of insulation
- Heat loss calculations
- Ceramic fibres and high emissivity coatings
- Refractories – classification and selection

5. INSULATION AND REFRACTORIES

Insulation-types and Application, Economic Thickness of Insulation, Heat Savings and Application Criteria, Cold Insulation, Refractory-types, Selection and Application of Refractories, Heat loss.

5.1 Purpose of Insulation

A thermal insulator is a poor conductor of heat and has a low thermal conductivity. Insulation is used in buildings and in manufacturing processes to prevent heat loss or heat gain. Although its primary purpose is an economic one, it also provides more accurate control of process temperatures and protection of personnel. It prevents condensation on cold surfaces and the resulting corrosion. Such materials are porous, containing large number of dormant air cells. Thermal insulation delivers the following benefits:

- Reduces over-all energy consumption
- Offers better process control by maintaining process temperature.
- Prevents corrosion by keeping the exposed surface of a refrigerated system above dew point
- Provides fire protection to equipment
- Absorbs vibration

5.2 Types and Application

The Insulation can be classified into three groups according to the temperature ranges for which they are used.

Low Temperature Insulations (up to 90^{o}C)

This range covers insulating materials for refrigerators, cold and hot water systems, storage tanks, etc. The commonly used materials are Cork, Wood, 85% magnesia, Mineral Fibers, Polyurethane and expanded Polystyrene, etc

Medium Temperature Insulations (90 – 325^{o}C)

Insulators in this range are used in low temperature, heating and steam raising equipment, steam lines, flue ducts etc. The types of materials used in this temperatures range include 85% Magnesia, Asbestos, Calcium Silicate and Mineral Fibers etc.

High Temperature Insulations (325^{o} C – above)

Typical uses of such materials are super heated steam system, oven dryer and furnaces etc. The most extensively used materials in this range are Asbestos, Calcium Silicate, Mineral Fibre, Mica and Vermiculite based insulation, Fireclay or Silica based insulation and Ceramic Fibre.

Insulation material

Insulation materials can also be classified into organic and inorganic types. Organic insulations are based on hydrocarbon polymers, which can be expanded to obtain high void structures

Example: Thermocol (Expanded Polystyrene) and Poly Urethane Form (PUF).

Inorganic insulation is based on Siliceous/Aluminous/Calcium materials in fibrous, granular or powder forms.

Example: Mineral wool, Calcium silicate etc.

Properties of common insulating materials are as under:

Calcium Silicate: Used in industrial process plant (Figure 5.1) piping where high service temperature and compressive strength are needed. Temperature ranges varies from 40 ^{o}C to 950 ^{o}C .

Glass mineral wool: These are available (Figure 5.1) in flexible forms, rigid slabs and preformed pipe work sections. Good for thermal and acoustic insulation for heating and chilling system pipelines. Temperature range of application is –10 to 500 ^{o}C

Thermocol: These are mainly used as cold insulation for piping and cold storage construction.

Calcium silicate sections	Glass Mineral wool

Figure 5.1

Expanded nitrite rubber: This is a flexible material that forms a closed cell integral vapour barrier. Originally developed for condensation control in refrigeration pipe work and chilled water lines; now-a-days also used for ducting insulation for air conditioning.

Rock mineral wool: This is available in a range of forms from light weight rolled products to heavy rigid slabs including preformed pipe sections. In addition to good thermal insulation properties, it can also provide acoustic insulation and is fire retardant.

Use of Moulded Insulation

Lagging materials can be obtained in bulk, in the form of moulded sections; semi - cylindrical for pipes, slabs for vessels, flanges, valves etc. The main advantage of the moulded sections is the ease of application and replacement when undertaking repairs for damaged lagging.

The thermal conductivity of a material is the heat loss per unit area per unit insulation thickness per unit temperature difference. The unit of measurement is W/m^2/m°C or W/m°C. The thermal conductivity of materials increases with temperature. So, thermal conductivity is always specified at the mean temperature (mean of hot and cold face temperatures) of the insulation material.

Thermal conductivities of typical hot insulation materials is given in Table 5.1

Table 5.1: Thermal Conductivity of Hot Insulation

Mean Temperature °C	Calcium Silicate	Resin bonded Mineral wool	Ceramic Fiber Blankets
100	-	0.04	-
200	0.07	0.06	0.06
300	0.08	0.08	0.07
400	0.08	0.11	0.09
700	-	-	0.17
1000	-	-	0.26
Specific heat(kJ/kg/°C)	0.96	0.921	1.07
	(at 40°C)	(at 20°C)	(at 980°C)
Service temp, (°C).	950	700	1425
Density kg/m^3	260	48 to 144	64 to 128

5.3 Calculation of Insulation Thickness

The most basic model for insulation on a pipe is shown in Figure 5.2.

r1 shows the outside radius of the pipe
r2 shows the radius of the pipe + insulation.

Heat loss from a surface is expressed as

H = h X A x (T_h-Ta)

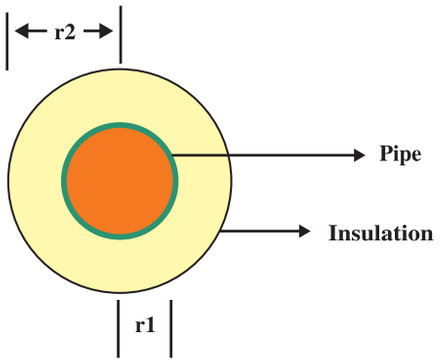

Figure 5.2 Pipe Insulation

Where,

h = Heat transfer coefficient, W/m^2-K
H = Heat loss, Watts
Ta = Average ambient temperature, ºC
Ts = Desired/actual insulation surface temperature, ºC
T_h = Hot surface temperature (for hot fluid piping), ºC & Cold surface temperature for cold fluids piping)

For horizontal pipes, heat transfer coefficient can be calculated by:

h = (A + 0.005 (T_h – Ta)) x 10 W/m^2-K

For vertical pipes,

h = (B + 0.009 (T_h – Ta)) x 10 W/m^2-K

Using the coefficients A, B as given below.

Surface	ε	A	B
Aluminium, bright rolled	0.05	0.25	0.27
Aluminium, oxidized	0.13	0.31	0.33
Steel	0.15	0.32	0.34
Galvanised sheet metal, dusty	0.44	0.53	0.55
Non metallic surfaces	0.95	0.85	0.87

$$T_m = \frac{(T_h + T_s)}{2}$$

k = Thermal conductivity of insulation at mean temperature of Tm, W/m-C
tk = Thickness of insulation, mm
r1 = Actual outer radius of pipe, mm
r2 = (r1 + tk)

R_s = Surface thermal resistance = $\frac{1}{h}$ °C-m²/W

R_I = Thermal resistance of insulation = $\frac{tk}{k}$ °C-m²/W

The heat flow from the pipe surface and the ambient can be expressed as follows

H = Heat flow, Watts

$$= \frac{(T_h - T_a)}{(R_I + R_s)} = \frac{(T_s - T_a)}{R_s}$$

From the above equation, and for a desired Ts, Rl can be calculated. From Rl and known value of thermal conductivity k, thickness of insulation can be calculated.

Equivalent thickness of insulation for pipe, $E_{tk} = (r_1 + t_k) \times \ln\left(\frac{(r_1 + t_k)}{r_1}\right)$

5.4 Economic Thickness of Insulation (ETI)

Insulation of any system means capital expenditure. Hence the most important factor in any insulation system is to analyse the thermal insulation with respect to cost. The effectiveness of insulation follows the law of decreasing returns. Hence, there is a definite economic limit to the amount of insulation, which is justified. An increased thickness is uneconomical and cannot be recovered through small heat savings. This limiting value is termed as economic thickness of insulation. An illustrative case is

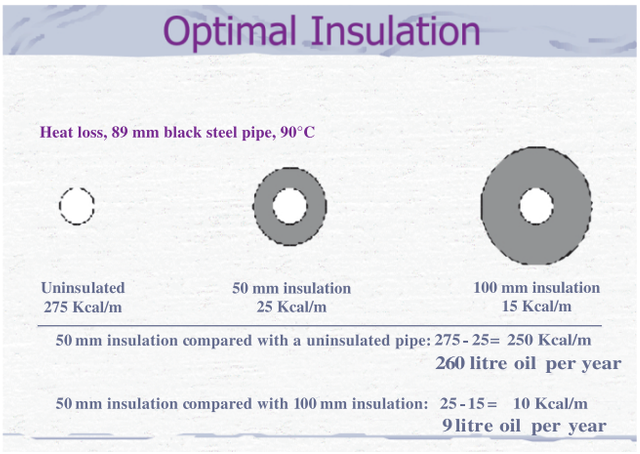

Figure 5.3 Illustration of Optimal Insulation

given in Figure 5.3. Each industry has different fuel cost and boiler efficiency. These values can be used for calculating economic thickness of insulation. This shows that thickness for a given set of circumstances results in the lowest overall cost of insulation and heat loss combined over a given period of time. The following Figure 5.4 illustrates the principle of economic thickness of insulation.

The simplest method of analysing whether you should use 1" or 2" or 3" insulation is by comparing the cost of energy losses with the cost of insulating the pipe. The insulation thickness for which the total cost is minimum is termed as economic thickness. Refer fig 5.4 The curve representing the total cost reduces initially and after reaching the economic thickness corresponding to the minimum cost, it increases.

The determination of economic thickness requires the attention to the following factors.

i. Cost of fuel
ii. Annual hours of operation
iii. Heat content of fuel
iv. Boiler efficiency
v. Operating surface temperature
vi. Pipe diameter/thickness of surface
vii. Estimated cost of insulation.
viii. Average exposure ambient still air temperature

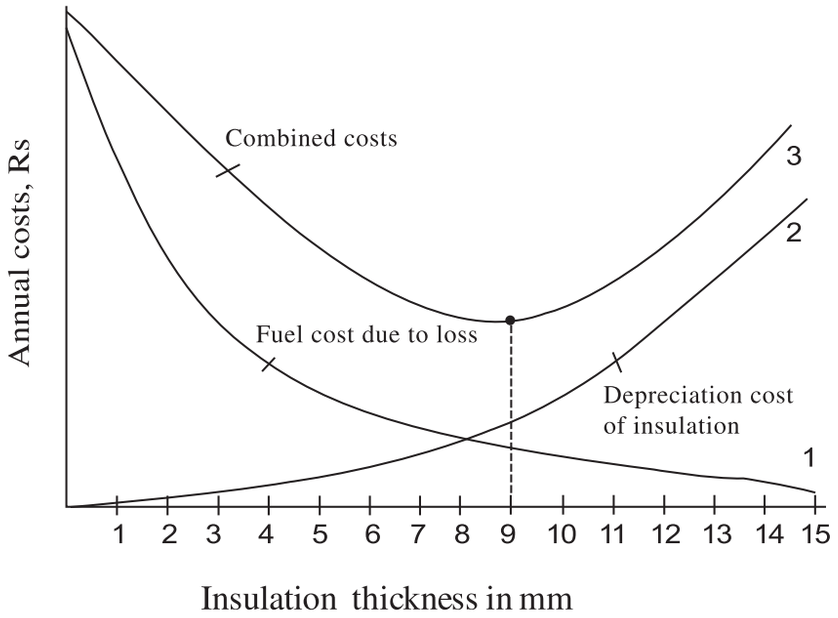

Figure 5.4 Determination of Economic Thickness of Insulation

Procedure for calculating Economic thickness of insulation

To explain the concept of economic thickness of insulation, we will use an example. (Refer Table 5.2) Consider an 8 bar steam pipeline of 6" dia having 50-meter length. We will evaluate the cost of energy losses when we use 1", 2" and 3" insulation to find out the most economic thickness.

A step-by-step procedure is given below.

1. Establish the bare pipe surface temperature, by measurement.
2. Note the dimensions such as diameter, length & surface area of the pipe section under consideration.
3. Assume an average ambient temperature. Here, we have taken 30°C.
4. Since we are doing the calculations for commercially available insulation thickness, some trial and error calculations will be required for deciding the surface temperature after putting insulation. To begin with assume a value between 55° & 65°C, which is a safe, touch temperature.
5. Select an insulation material, with known thermal conductivity values in the mean insulation temperature range. Here the mean temperature is 111°C. and the value of k = 0.044 W/m^{2}°C for mineral wool.
6. Calculate surface heat transfer coefficients of bare and insulated surfaces, using equations discussed previously. Calculate the thermal resistance and thickness of insulation.
7. Select r2 such that the equivalent thickness of insulation of pipe equals to the insulation thickness estimated in step 6. From this value, calculate the radial thickness of pipe insulation = r2-r1
8. Adjust the desired surface temperature values so that the thickness of insulation is close to the standard value of 1" (25.4 mm).
9. Estimate the surface area of the pipe with different insulation thickness and calculate the total heat loss from the surfaces using heat transfer coefficient, temperature difference between pipe surface and ambient.
10. Estimate the cost of energy losses in the 3 scenarios. Calculate the Net Present Value of the future energy costs during an insulation life of typically 5 years.
11. Find out the total cost of putting insulation on the pipe (material + labor cost)
12. Calculate the total cost of energy costs and insulation for 3 situations.
13 Insulation thickness corresponding to the lowest total cost will be the economic thickness of insulation.

Table 5.2 Example for Economic Insulation Thickness

		Insulation thickness		
Description	Unit	1"	2"	3"
Length of pipe, L	m	50	50	50
Bare Pipe outer diameter, d1	mm	168	168	168
Bare pipe surface area, A	m^2	26.38	26.38	26.38
Ambient Temperature, Ta :	°C	30	30	30
Bare Pipe Wall Temperature, Th:	°C	160	160	160
Desired Wall Temperature With Insulation, Tc :	°C	62	48	43
Material of Insulation :		Mineral Wool		
Mean Temperature of Insulation, Tm = (Th+Tc)/2 :	°C	111	104	101.5
Sp.Conductivity of Insulation Material, k (from catalogue) :	W/m°C	0.044	0.042	0.04
Surface Emissivity of bare pipe:		0.95	0.95	0.95
Surface emissivity of insulation cladding(typically Al)		0.13	0.13	0.13

Calculations

Surface Heat Transfer Coefficient of Hot Bare Surface, h :(0.85+ 0.005 (Th – Ta)) x 10	W/m^2°C	15	15	15
Surface Heat Transfer Coefficient After Insulation, h' = (0.31+ 0.005 (Tc – Ta)) x 10	W/m^2°C	4.7	4	3.75
Thermal Resistance, R_{th} = (Th-Tc)/[h' x (Tc-Ta)] :	°C-m^2/W	0.7	1.6	2.4
Thickness of Insulation, t = k x R_{th} :(if surface was flat)	mm	28.7	65.3	96.0
r_1=outer diameter/2 =	mm	84	84	84
$t_{eq} = r_2 \times \ln(r_2/r_1)$ = (select r_2 so that t_{eq} = t)	mm	28.7	65.3	106.3
Outer radius of insulation , r_2=	mm	109.2	135.9	161.9
Thickness of insulation	mm	**25.2**	**51.9**	**77.9**
Insulated pipe Area , A :	m^2	34.29	42.66	50.85
Total Losses From Bare Surface, Q = h x A x (Th-Ta) :	kW	51.4	51.4	51.4
Total Loss From Insulated Surface, Q' = h' x A' x (Tc-Ta) :	kW	5.16	3.07	2.48
Power Saved by Providing Insulation, P = Q - Q' :	kW	46.3	48.4	49.0
Annual Working Hours, n :	Hrs	8000	8000	8000
Energy Saving After Providing Insulation, E = P x n :	kWh/year	370203	386892	391634

Economics

Steam cost,	Rs/kg	0.70	0.70	0.70
Heat Energy Cost, p :	Rs./kWh	1.11	1.11	1.11
Annual Monetary Saving, S = E x p :	Rs.	412708	431313	436599
Discount factor for calculating NPV of cost of energy loss	%	15%	15%	15%
Cost of insulation (material + labor)	Rs/m	450	700	1100
Total cost of insulation	Rs/m	22500	35000	55000
Annual Cost of energy loss	Rs/year	46000	27395	22109
NPV of annual cost of energy losses for 5 years	Rs	154198	91832	74112
Total cost (insulation & NPV of heat loss)	Rs	**176698**	**126832**	**129112**

Note that the total cost is lower when using 2" insulation, hence 2"size is the economic insulation thickness.

5.5 Simplified Formula for Heat Loss Calculation

Various charts, graphs and references are available for heat loss computation. The surface heat loss can be computed with the help of a simple relation as given below. This equation can be used up to 200 ^{0}C surface temperature. Factors like wind velocities, conductivity of insulating material etc has not been considered in the equation.

$$S = [10+(Ts-Ta)/20] \times (Ts-Ta)$$

Where,

S = Surface heat loss in kcal/hr m^2
Ts = Hot surface temperature in °C
Ta = Ambient temperature in °C

Total heat loss/hr (Hs) = S x A

Where,

A is the surface area in m^2

Based on the cost of heat energy, the quantification of heat loss in Rupees can be worked out as under:

$$\text{Equivalent fuel loss } (H_f) \text{ (kg/Yr)} = \frac{\text{Hs x Yearly hours of operation}}{\text{GCV x } \acute{\eta}_b}$$

$$\text{Annual heat loss in monetary terms (Rs.)} = H_f \text{ x Fuel cost (Rs./kg)}$$

Where

GCV = Gross Calorific value of fuel kcal/kg

η_b = Boiler efficiency in %

Case Example

Steam pipeline 100 mm diameter is not insulated for 100 metre length supplying steam at 10 kg/cm^2 to the equipment. Find out the fuel savings if it is properly insulated with 65 mm insulating material.

Assumptions:

Boiler efficiency – 80%
Fuel Oil cost - Rs.15000/tonne
Surface temperature without insulation – 170°C
Surface temperature after insulation – 65°C
Ambient temperature – 25°C

Existing Heat Loss

$S = [10+ (T_s-T_a)/20] \times (T_s-T_a)$
$T_s = 170^oC$
$T_a = 25^oC$
$S = [10+(170-25)/20] \times (170-25) = 2500$ kcal/hr-m^2
S1 = S =Existing heat loss (2500 kcal/hr-m^2)

Modified System

After insulating with 65 mm glass wool with aluminum cladding the hot face temperature will be 65 °C

Ts – 65 °C
Ta – 25 °C

Substituting these values

S = [10+(65-25)/20] x (65-25)
= 480 kcal/hr m^2

S2 = S =Existing heat loss (480 kcal/hr-m^2)

Fuel Savings Calculation

Pipe dimension	=	ϕ 100 mm & 100 m length
Surface area existing (A1)	=	3.14 x 0.1 x 100 = 31.4 m^2
Surface area after insulation (A2)	=	3.14 x 0.23 x 100 = 72.2 m^2
Total heat loss in existing system (S1 x A1)	=	2500 x 31.4 = 78500 kCal/hr
Total heat loss in Modified system (S2 x A2)	=	480 x 72.2 = 34656 kCal/hr
Reduction in heat loss	=	78500 – 34656 = 43844 kCal/hr
No. of hours operation in a year	=	8400
Total heat loss (kCal/yr)	=	43844 x 8400 = 368289600
Calorific value of fuel oil	=	10300 kCal/kg
Boiler efficiency	=	80%
Price of fuel oil	=	Rs.35000/Tonne
Yearly fuel oil savings	=	368289600/10300 x 0.8
	=	44695 kg//year

5.6 Cold Insulation

Cold Insulation Features

Cold Insulation should be considered and where operating temperature are below ambient where protection is required against heat gain, condensation or freezing. Condensation will occur whenever moist air comes into contact with the surface that is at a temperature lower than the due point of the vapour. In addition, heat gained by uninsulated chilled water lines can adversely affect the efficiency of the cooling system. The most important characteristics of a suitable Cold insulation material have following features:

- Low thermal conductivity
- High water resistance, and
- Durability at low temperature

Other properties like easy workability, negligible capillary absorption should also be taken into consideration while making a selection. The insulation system is only as good as its vapour barrier and the care with which it is installed.

Material Selection for Cold Insulation

Selection of insulation materials should be carefully considered where the possibility of steam purging of the equipment is required or for other reasons which may cause the temperature to be increased to a level that exceeds the maximum limiting temperature of the insulation materials, i.e., material then deteriorate. Examples of cold insulation include Urethane Foam, Expanded Polystyrene, Resin bonded glass wool, Resin Bonded Glass wool, and Phenolic Foam.

Economics of Cold Insulation:

Unlike hot insulation system, the concern area in Cold Insulation is the heat gain into the refrigerated space, which leads to increase in the refrigeration load (TR) & energy consumption as a consequence. The cost of heat gain can thus be assessed & evaluated against cost of additional cold insulation thickness, to optimize overall energy consumption & cost in refrigeration system.

5.7 Refractories

Any material can be described as 'refractory,' if it can with stand the action of abrasive or corrosive solids, liquids or gases at high temperatures. Examples are: fireclay, alumina, magnesite, chrome magnesite, dolomite etc. Refractory materials are made in varying combinations and shapes and for different requirements of high temperature processes carried out in metal extraction, cement, glassmaking, manufacturing, ceramic etc.

The general requirements of a refractory material can be summed up as :

- Ability to withstand high temperatures.
- Ability to withstand sudden changes of temperatures.
- Ability to withstand action of molten metal slag, glass, hot gases, etc.
- Ability to withstand load at service conditions.
- Ability to withstand load and abrasive forces.
- Low coefficient of thermal expansion.
- Should be able to conserve heat.
- Should not contaminate the material with which it comes into contact.

5.8 Properties of Refractories

Some of the important properties of refractories are:

Size : The size and shape of the refractories is a part of the design feature. It is an important feature in design since it affects the stability of any structure. Accuracy and size is extremely important to enable proper fitting of the refractory shape and to minimize the thickness and joints in construction.

Bulk density: A useful property of refractories is bulk density, which defines the material present in a given volume. An increase in bulk density of a given refractory increases its volume stability, its heat capacity, as well as resistance to slag penetration.

Porosity: The apparent porosity is a measure of the volume of the open pores, into which a liquid can penetrate, as a percentage of the total volume. This is an important property in cases where the refractory is in contact with molten charge and slags. A low apparent porosity is desirable since it would prevent easy penetration of the refractory size and continuity of pores will have important influences on refractory behaviour. A large number of small pores is generally preferable to an equivalent number of large pores.

Cold crushing strength: The cold crushing strength, which is considered by some to be of doubtful relevance as a useful property, other than that it reveals little more than the ability to withstand the rigors of transport, can be used as a useful indicator to the adequacy of firing and abrasion resistance in consonance with other properties such as bulk density and porosity.

Pyrometric cone equivalent (PCE): Temperature at which a refractory will deform under its own weight is known as its softening temperature which is indicated by PCE. Refractories, due to their chemical complexity, melt progressively over a range of temperature. Hence refractoriness or fusion point is ideally assessed by the cone fusion method. The equivalent standard cone which melts to the same extent as the test cone is known as the pyrometric cone equivalent.

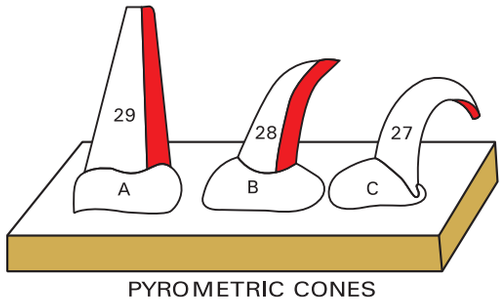

Figure 5.5

Thus in the Figure 5.5 refractoriness of Sample A is much higher than B and C. The pyrometric cone equivalent indicates only the softening temperature. But, in service the refractory is subjected to loads which would deform the refrectory at a much lower temperature than that indicated by PCE. With change in the environmental conditions, such as reducing atmosphere, the P.C.E. value changes drastically.

Refractoriness under load (RUL): The refractoriness under load test (RUL test) gives an indication of the temperature at which the bricks will collapse, in service conditions with similar load.

Creep at high temperature: Creep is a time dependent property which determines the deformation in a given time and at a given temperature by a material under stress.

Volume stability, expansion, and shrinkage at high temperatures: The contraction or expansion of the refractories can take place during service. Such permanent changes in dimensions may be due to several operational factors.

Reversible thermal expansion: Any material when heated, expands, and contracts on cooling. The reversible thermal expansion is a reflection on the phase transformations that occur during heating and cooling.

Thermal conductivity: Thermal conductivity depends upon the chemical and mineralogical compositions as well as the glassy phase contained in the refractory and the application temperature. The conductivity usually changes with rise in temperature. In cases where heat transfer is required though the brick work, for example in recuperators, regenerators, muffles, etc. the refractory should have high conductivity. Low thermal conductivity is desirable for conservation of heat by providing adequate insulation.

The provisions for back-up insulation, conserves heat but at the same time it increases the hot face temperature and hence the demand on the refractory quality increases.

Light weight refractories of low thermal conductivity find wider applications in the moderately low temperature heat treatment furnaces, where its primary function is usually conservation of energy. It is more so in case of batch type furnaces where the low heat capacity of the refractory structure would minimize the heat storage during the intermittent heating and cooling cycles.

5.9 Classification of Refractories

Refractories can be classified on the basis of chemical composition and use and methods of manufacture is given in Table 5.3.

Table 5.3 Classification Based on Chemical Composition

Classification Based on Chemical Composition	Examples
ACID---which readily combines with bases.	Silica, Semisilica, Aluminosilicate.
BASIC---which consists mainly of metallic oxides which resist the action of bases.	Magnesite, chromemagnesite, Dolomite.
NEUTRAL---which doesn't combine; neither with acids nor bases.	Chrome, Pure. Alumina
Special	Carbon, Silicon Carbide, Zirconia.
Classification based on end use	Blast furnace Casting Pit
Classification based on method of manufacture	- Dry Press Process - Fused Cast - Hand Moulded - Formed Normal, fired or Chemically bonded.) - Unformed (Monolithics-plastics, Ramming Mass, Gunning Castable, Spraying.)

5.10 Typical Refractories in Industrial Use

Depending on the area of application such as boilers, furnaces, kilns, ovens etc, temperatures and atmospheres encountered different types of refractories are used. Typical installations of refractories are shown in Figure 5.6

Refractory lining of a furnace arch

Refractory walls of a furnace interior with burner blocks

Figure 5.6

Fireclay Refractories

Fireclay refractories, such as firebricks, siliceous fireclays and aluminous clay refractories consist of aluminium silicates with various amounts of silica ranging from SiO_2 content of less than 78% and containing less than 44% of Al_2O_3.

Table 5.4 shows that as the quantity of impurities increases and the amount of Al_2O_3 decreases, the melting point of fireclay brick decreases. Owing to its relative cheapness and widespread location of the raw materials used to manufacture firebricks, this material finds use in most furnaces, kilns, stoves, etc.

Table 5.4 Properties of Typical Fireclay Bricks

Brick	Percent SiO_2	Percent Al_2O_3	Other Constituents	PCE ^0C
Super Duty	49-53	40-44	5-7	1745-1760
High Duty	50-80	35-40	5-9	1690-1745
Intermediate	60-70	26-36	5-9	1640-1680
High Duty (Siliceous)	65-80	18-30	3-8	1620-1680
Low Duty	60-70	23-33	6-10	1520-1595

Firebrick is the most common form of refractory material. It is used extensively in the iron and steel industry, nonferrous metallurgy, glass industry, pottery kilns, cement industry, and by many others.

High Alumina Refractories

Alumino silicate refractories containing more than 45% alumina are generally termed as high alumina materials. The alumina concentration ranges from 45 to 100%. The refractoriness of high alumina

refractories increases with increase in alumina percentage. The applications of high alumina refractories includes the hearth and shaft of blast furnaces, ceramic kilns, cement kilns, glass tanks and crucibles for melting a wide range of metals.

Silica Brick

Silica brick (or Dinas) is a refractory material containing at least 93% SiO_2. The raw material is quality rocks. Various grades of silica brick have found extensive use in the iron and steel melting furnaces. In addition to high fusion point multi-type refractories, the other important properties are their high resistance to thermal shock (spalling) and their high refractoriness. It finds typical use in glass making and steel industry.

The outstanding property of silica brick is that it does not begin to soften under high loads until its fusion point is approached. This behaviour contrasts with that of many other refractories, for example alumino silicate materials, which begin to fuse and creep at temperatures considerably lower than their fusion points. Other advantages are flux and stag resistance, volume stability and high spalling resistance.

Magnesite

Magnesite refractories are chemically basic materials, containing at least 85% magnesium oxide. They are made from naturally occurring magnesite ($MgCO_3$). The properties of magnesite refractories depend on the concentration of silicate bond at the operating temperatures. Good quality magnesite usually results from a CaO-SiO_2 ratio of less than 2 with a minimum ferrite concentration, particularly if the furnaces lined with the refractory operate in oxidizing and reducing conditions. The slag resistance is very high particularly to lime and iron rich slags.

Chromite Refractories

Chrome-magnesite refractories are made in a wide range of qualities and are used for building the critical parts of high temperature furnaces. These materials can withstand corrosive slags and gases and have high refractoriness.

The magnesite-chromite refractories on the other hand are suitable for service at the highest temperatures and in contact with the most basic slags used in steel melting. Magnesite-chromite usually ahs a better spalling resistance than chrome-magnesite.

Zirconia Refractories

Zirconium dioxide (ZrO_2) is a polymorphic, material. There are certain difficulties in its usage and fabrication as a refractory material. It is essential to stabilize it before application as a refractory. This is achieved by incorporating small quantities of calcium, magnesium and cerium oxide, etc. Its properties depend mainly on the degree of stabilization and quantity of stabilizer as well as the quality of the original raw material. Zirconia refractories have a very high strength at room temperature which is

maintained upto temperatures as high as 1500^{0}C. They are, therefore, useful as high temperature constructional materials for furnaces and kilns. The thermal conductivity, of zirconium dioxide is found to be much lower than that of most other refractories and the material is therefore used as a high temperature insulating refractory.

Since Zirconia exhibits very low thermal losses and does not react readily with liquid metals, it is particularly useful for making refractory crucibles and other vessels for metallurgical purposes. Zirconia is a useful refractory material for glass furnaces primarily since it is not easily wetted by molten glasses and because of its low reaction with them.

Oxide Refractories (Alumina)

Alumina refractory materials which consist of aluminium oxide with little traces of impurities are often known as pure alumina. Alumina is one of the most chemically stable oxides known. It is mechanically very strong, insoluble in water and super heated steam, and in most inorganic acids and alkalies. Its properties make it suitable for the shaping of crucibles for fusing sodium carbonate, sodium hydroxide and sodium peroxide. It has a high resistance in oxidizing and reducing atmosphere. Alumina is extensively used in heat processing industries. Highly porous alumina is used for lining furnaces operating up to 1850^{0}C.

Monolithics

Monolithic refractories (single piece cast in the shape of equipment such as one for a ladle shown in Figure 5.7) are replacing the conventional type fired refractories at a much faster rate in many applications including those of industrial furnaces. The main advantages being:

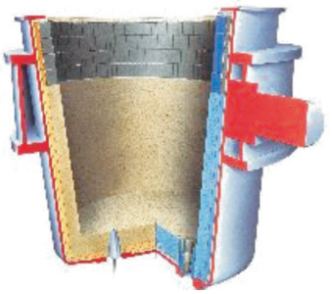

Figure 5.7 Monolithic Lining for a Ladle

- It eliminates joints which is an inherent weakness
- Method of application is faster and skilled measures in large number are not required
- Transportation and handling are simple
- Offers better scope to reduce downtime for repairs
- Offers considerable scope to reduce inventory and eliminate special shapes
- It is a heat saver
- Has better spalling resistance
- Has greater volume stability

Various means are employed in the placement of monolithics like ramming, casting, gunniting, spraying, sand slinging, etc. Ramming masses are used mostly in cold applications where proper consolidation of the material is important. The same practice can be adopted with both air setting and heat setting materials. Proper ramming tools need to be selected.

Castables by name implies a material of hydraulic setting in nature. Calcium aluminate cement being the binder, it will have to be stored properly to prevent moisture absorption. Further its strength starts deteriorating after a period of 6 to 12 months.

Insulating materials

Insulating materials greatly reduce the heat losses through walls. Insulation is effected by providing a layer of material having a low heat conductivity between the internal hot surface of a furnace and the external surface, thus causing the temperature of the external surface reduced.

The insulating materials may be classified into the following groups

- Insulating bricks
- Insulating Castables
- Ceramic fibre
- Calcium silicate
- Ceramic coating

Insulating materials owe their low conductivity to their pores while their heat capacity depends on the bulk density and specific heat. Structure of air insulating material consists of minute pores filled with air which have in themselves very low thermal conductivity, excessive heat affects all insulation material adversely, but the temperatures to which the various materials can be heated before this adverse effect occurs differ widely. Clearly, therefore, the choice of an insulating material must depend upon its effectiveness to resist heat conductivity and upon the temperature that it will withstand.

One of the most widely used insulating materials is diatomite, also known as kiesel guhr which is made up of a mass of skeletons of minute aquatic plants deposited thousands of years ago on the beds of seas and lakes. Chemically this consists of silica contaminated with clay and organic matter. Wide ranges of insulating refractories with wide combinations of properties are now available.

The important physical properties of some insulating refractories are shown in the Table 5.5.

Table 5.5 Physical Properties of Insulating Refractories

Type	Thermal Conductivity at 400^{0}C	Max. safe Temperature ^{0}C	Cold Crushing Strength kg/cm^2	Porosity %	Bulk density kg/m^3
Diatomite Solid Grade	0.025	1000	270	52	1090
Diatomite Porous Grade	0.014	800	110	77	540
Clay	0.030	1500	260	68	560
High Aluminia	0.028	1500-1600	300	66	910
Silica	0.040	1400	400	65	830

Castables and Concretes

Monolithic linings and furnace sections can be built up by casting refractory insulating concretes, and by stamping into place certain light weight aggregates suitably bonded. Other applications include the formation of the bases of tunnel kiln cars used in the ceramic industry. The ingredients are similar to those used for making piece refractories, except that concretes contain some kind of cement, either Portland or high-alumina cement.

Ceramic Fibre

Ceramic fibre (Figure 5.8) is a low thermal mass insulation material, which has revolutionalised the furnace design lining systems.

Figure 5.8: Ceramic Fibre Insulation

Ceramic fibre is an alumino silicate material manufactured by blending and melting alumina and silica at temperature of 1800-2000°C and breaking the molten stream by blowing compressed air or dropping the melt on spinning disc to form loose or bulk ceramic fibre. The bulk fibre is converted to various products including blanket, strips, veneering and anchored modules, paper, vacuum formed boards and shapes, rope, wet felt, mastic cement etc. for insulation applications. The properties of ceramic fibre in comparison with with conventional refractories is given in Table 5.6

Table 5.6 Typical Refractory Properties

Property	High Thermal Mass (High Density Refractories)	Low Thermal Mass (Ceramic Fibre)
Thermal Conductivity, W/m K	1.2	0.3
Specific Heat, J/kg K	1000	1000
Density, kg/m^3	2300	130

Fibres are usually produced in two temperature grades based on Al_2O_3 content. A recent addition is Zr O_2 added alumino silicate fibre, which helps to reduce shrinkage levels thereby rating the fibre for higher temperatures. Continuous recommended operating temperature for fibres are given in the following Table 5.7.

Table 5.7 Continuous Recommended Operating Temperature for Fibres

	$A1_2O_3$	SiO_2	ZrO_2
1150°C	43 - 47 %	53 - 57 %	-
1250°C	52 - 56 %	44 - 48 %	-
1325°C	33 - 35 %	47 - 50 %	17 - 20 %

These fibres are generally produced in bulk wool form and needled into blanket mass of various densities ranging from 64 to 190 kg/m^3. Converted products and over 40 different forms are made from blankets to suit various requirements.

Important Properties of Ceramic Fibre

The characteristics of ceramic fibres are a remarkable combination of the properties of refractories and traditional insulation material.

1. Lower Thermal Conductivity

The low thermal conductivity – 0.1 kcal/m hour deg C at 600^oC for 128 kg/m^3 density blanket – allows construction of thinner linings with the same thermal efficiency as that of conventional refractories. Hence, for the same outer envelope dimension the furnace volume is much higher. It is 40 % more effective than good quality insulation brick and 2.5 times better than asbestos product. Insulating property of ceramic fibre is better than calcium silicate product.

2. Light Weight

Average density of ceramic fibre is 96 kg/m^3. It is one tenth of the weight of insulating brick and one third that of asbestos / calcium silicate boards. For new furnaces structural supports can be reduced by 40%.

3. Lower Heat Storage

Ceramic fibre linings absorb less heat because of lower density. Furnace can be heated and cooled at faster rates. Typically the heat stored in a ceramic fibre lining system is in the range of 2700 - 4050 kcal/m^2 (1000 – 1500 Btu/Ft^2) as compared to 54200-493900 kcal/m^2 (20000 – 250000 Btu/Ft^2) for conventionally lined system.

4. Thermal Shock Resistant

Ceramic fibre lining resist thermal shock due to their resilient matrix. Also faster heat up and cool down cycles are possible thereby improving furnace availability and productivity.

5. Chemical Resistance

Ceramic fibre resist most of the chemical attack and is unaffected by hydrocarbons, water and steam present in flue gases.

6. Mechanical Resilience

This property permits fibre lined furnaces to be shop fabricated and shipped to site in assembled form without fear of damage.

7. Low Installation Cost

No special skills are required as application practices are standardised. Fibre linings require no dry out or curing times and can be heated to the capacity of the burners after installation is completed without concern for cracking or spalling.

8. Simple Maintenance

In case of physical damage the defective section can be quickly removed and a replacement piece added. Whole panel sections can be prefabricated for fast installation with minimal down time.

9. Ease of Handling

All product forms are easily handled and most can be quickly cut with a knife or scissors. Vacuum formed products may require cutting with a band saw.

10. Thermal Efficiency

The low thermal conductivity of ceramic fibre can be advantageously made use of by the lesser lining thickness and reduced furnace volume. The fast response of ceramic fibre lined furnace also allows for more accurate control and uniform temperature distribution within the furnace.

The other advantages offered by ceramic fibre are summarized below:

- Light weight furnace
- Simple steel fabrication work
- Low down time
- Increased productivity
- Additional capacity
- Low maintenance cost
- Longer service life
- Higher thermal efficiency
- Faster response

High Emissivity Coatings

Emissivity, the measure of a material's ability to both absorb and radiate heat, has been considered by engineers as being an inherent physical property which like density, specific heat and thermal conductivity, is not readily amenable to change. However, the development of high emissivity coatings now allows the surface emissivity of materials to be increased, with resultant benefits in heat transfer efficiency and in the service life of heat transfer components. High emissivity coatings are applied in the interior surface of furnaces. The Figure 5.9 shows emissivity of various insulating materials including high emissivity coatings. High emissivity coating shows a constant value over varying process temperatures.

The application of high-emissivity coatings in furnace chambers promotes rapid and efficient transfer of heat, uniform heating, and extended life of refractories and metallic components such as radiant tubes and heating elements. For intermittent furnaces or where rapid heating is required, use of such coatings was found to reduce fuel or power to tune of 25-45%. Other benefits are temperature uniformity and increased refractory life.

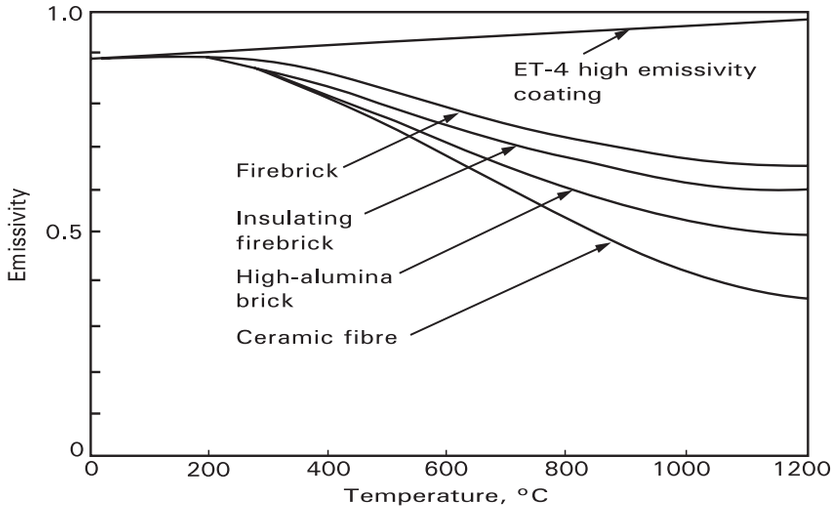

Figure 5.9 Emissivity of Refractory Materials at Different temperatures

Furnaces, which operate at high temperature, have emissivities of 0.3. By using high emissivity coatings this can go upto 0.8 thus effectively increasing the radiative heat transfer.

5.11 Selection of Refractories

The selection of refractories for any particular application is made with a view to achieve the best performance of the equipment furnace, kiln or boiler and depends on their properties. Further, the choice of a refractory material for a given application will be determined by the type of furnace or heating unit and the prevailing conditions e.g. the gaseous atmosphere, the presence of slags, the type of metal charge etc. It is, therefore, clear that temperature is by no means the only criterion for selection of refractories.

Any furnace designer or industry should have a clear idea about the service conditions which the refractory is required to face. The furnace manufacturers or users have to consider the following points, before selecting a refractory.

- Area of application.
- Working temperatures.
- Extent of abrasion and impact.
- Structural load of the furnace.
- Stress due to temperature gradient in the structures and temperature fluctuations.
- Chemical compatibility to the furnace environment.
- Heat transfer and fuel conservation
- Cost considerations.

It is therefore, essential to have an objective evaluation of the above conditions. A proper assessment of the desired properties would provide guidelines for selection of the proper refractory materials.

It would be important to mention here that the furnace manufacturer or a user is also concerned with the conservation of energy. Fuel can be saved in two ways: either by insulation or by faster working. Both these methods give low energy cost per tonne of product.

5.12 Heat Losses from Furnace Walls

In furnaces and kilns, heat losses from furnace walls, affect the fuel economy substantially. The extent of wall losses depends on:

i) Emissivity of walls;
ii) Conductivity of refractories;
iii) Wall thickness;
iv) Whether furnace or kiln is operated continuously or intermittently.

Heat losses can be reduced by increasing the wall thickness, or through the application of insulating bricks. Outside wall temperature and heat losses for a composite wall of a certain thickness of firebrick and insulation brick are much lower due to lesser conductivity of insulating brick as compared to a refractory brick.

In the case of batch furnace operation, operating periods ("on") alternate with idle periods ("off"). During the off period, the heat stored in the refractories in the on-period is gradually dissipated, mainly through radiation and convection from the cold face. In addition, some heat is obstructed by air flowing through the furnace. Dissipation of stored heat is a loss, because the lost heat is at least in part again imparted to the refractories during the next "on" period, thus expending fuel to generate the heat. If a furnace is operated 24 hr. every third day, practically all of the heat stored in the refractories is lost.

Solved Example:

A steam pipeline of 250 mm outer diameter & 100 meters long is insulated with 150 mm Mineral wool insulation. As an energy conservation measure, the management has upgraded the existing Mineral wool insulation with efficient calcium silicate insulation. Calculate the economics in terms of payback if the insulation is upgraded at a cost of 20 lakhs.

Given:

Operating hours	: 8000
Boiler efficiency	: 87 %
Fuel Oil Cost	: Rs. 45,000 per ton
GCV of the fuel	: 10,200 kcal/kg
Thickness of Mineral wool insulation	: 150 mm
Thickness of Calcium Silicate insulation	: 100 mm
Surface temperature with Mineral wool insulation	: 70 °C
Surface temperature with Calcium silicate insulation	: 55 °C
Ambient temperature	: 30 °C

Ans:

Heat loss with Mineral wool insulation	= {10 + (Ts –Ta)/20} x (Ts – Ta) = (10 + (70 – 30)/20 } x (70 – 30) = 480 kcal/hr per Square meter
Heat loss with Calcium silicate insulation	= (10 + (55 – 30)/20 } x (55 – 30) = 281.25 kcal/hr – Sq. m
Surface Area with Mineral wool	= 3.14 D L = 3.14 x 0.550 x 100 = 172.7 Square meter
Surface Area with Calcium Silicate	= 3.14 x 0.450 x 100 = 141.3 Square meter
Heat loss with Mineral wool	= 480 x 172.7 = 82896 kcal/hr
Heat loss with Calcium silicate	= 281.25 x 141.3 = 39741 kcal/hr
Annual savings	= $\frac{(82,896 - 39,741) \times 8,000 \times 45,000}{10,200 \times 1000 \times 0.87}$ = Rs. 17.5 Lakhs / year
Payback period	= 20 / 17.5 = 1.14 years = 13.7 months

QUESTIONS	
Objective Type Questions	
1.	Which of the following is not a property of ceramic fibre insulation? a) low thermal conductivity b) light weight c) high heat capacity d) thermal shock resistant
2.	____ indicates the softening temperature of the refractory a) refractoriness under load (RUL) b) cold crushing strength c) pyrometric cone equivalent (PCE) d) porosity
3.	An increase in bulk density of a refractory increases its a) volume stability b) heat capacity c) resistance to slag penetration d) all of the above
4.	The insulation material used for medium temperature application is a) calcium silicate b) polyurethane c) wood d) mica
5.	The heat loss rate from a surface is expressed in a) Joules b) W/m^2 °C c) Watt/sec d) kcal/m^2/hr
6.	The unit of thermal conductivity is a) W / m^2 ^oC b) W ^oC / m^2 c) W / m ^oC d) W ^oC / m
7.	Which of the following is not a property of ceramic fibre ? a) low thermal conductivity b) light weight c) high heat capacity d) thermal shock resistant
8.	The emissivity of ceramic coatings used in furnace a) decreases with increase in furnace temperature b) Increases with increase in furnace temperature c) remains constant d) decreases with increase in furnace pressure
9.	Glass mineral wool can be applied for temperature range application upto a) 950 ^oC b) 500 ^oC c) 1200 ^oC d) 750 ^oC
10.	Insulation used for temperatures more than 350°C is a) polyurethane b) polystyrene c) calcium silicate d) wood
Short Type Questions	
S-1	Draw the appropriate curves and mark the point of most economical insulation thickness in a X – Y coordinate system where the Y-axis represents annual costs and X-axis represents the insulation thickness of a furnace.
S-2	List some applications using ceramic fibre.
S-3	In selection of a refractory which physical, chemical and other properties of a refractory are important?
S-4	Describe any five properties of ceramic fibre from the point of view of their uses in furnace.
S-5	Name five factors on which the furnace wall losses depend on

Long Type Questions

L-1 A steam pipeline of 100 mm outer diameter is not insulated for 100 meters and supplying steam at 10 kg/cm^2. Find out the annual fuel savings if the line is properly insulated with 65 mm insulating material. Assume 7000 hours/year of operation.

Given:

Boiler efficiency	:	90%
Fuel Oil cost	:	Rs.13,000/tonne
Calorific Value of fuel oil	:	10,300 kcal/kg
Surface temperature without insulation	:	180^{o}C
Surface temperature after insulation	:	75^{o}C
Ambient temperature	:	30^{o}C

L-2 The insulation of a steam pipeline is to be upgraded. With the following data calculate the simple payback period for the insulation upgradation project.

Length of the steam pipeline	100 m
Bare pipe external diameter	100 mm
Heat loss from the pipe with existing 25 mm insulation thickness	2000 kcal/m^{2}hr
Thickness of insulation to be added	25 mm
Heat loss after insulation upgradation	400 kcal/m^2/hr
Boiler efficiency	75%
GCV of coal	4000 kcal/kg
Annual operating hours	8000 hrs
Cost of coal	Rs. 2000/Ton
Investment for insulation upgradation	Rs. 4 lakhs

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Chapter 6

FBC Boilers

Learning Objectives

In this chapter you will learn about

- Principles of fluidised bed combustion
- Types of fluidised bed combustion boilers
- Retrofitting aspects
- Advantages of fluidised bed combustion
- Application considerations with biomass FBC boiler

6. FBC BOILERS

Introduction, Mechanism of fluidized bed combustion, Types of FBC boilers, Retrofitting of FBC Systems to Conventional Boilers, Advantages of Fluidised Bed Combustion Boilers, Application Considerations with Biomass FBC Boilers, Retrofit of FBC Systems.

6.1 Introduction

The major portion of the coal available in India is of low quality, high ash content and low calorific value. The traditional grate fuel firing systems have got limitations and are techno-economically unviable to meet the challenges of future. Fluidized bed combustion has emerged as a viable alternative and has significant advantages over conventional firing system and offers multiple benefits – compact boiler design, fuel flexibility, higher combustion efficiency and reduced emission of noxious pollutants such as SOx and NOx. The fuels burnt in these boilers include coal, washery rejects, rice husk, bagasse & other agricultural wastes. The fluidized bed boilers have a wide capacity range- 0.5 T/hr. to over 100 T/hr.

6.2 Mechanism of Fluidized Bed Combustion

6.2.1 Primary Air

The primary air is the air supplied to a combustion system that makes first contact with fuel. The primary air ensures instant combustion as the fuel enters the furnace. In case of solid fuels, primary air is introduced under the grate and passes through the bed of fuel to burn the fixed carbon content.

6.2.2 Secondary Air

The secondary air provides enough oxygen to complete the combustion in the furnace. In case of solid fuels the secondary air is introduced over the grate (fuel bed) to react with volatile matter released from coal during the initial burning and help in completing the combustion. It is injected into the furnace of the boiler under pressure to create turbulence above the burning fuel which helps to ensure good mixing with gases produced in the combustion process and thereby completing the combustion.

6.2.3 Tertiary Air

The injection of air into a furnace in addition to primary and secondary air is called tertiary air. It creates further turbulence and burns the residual volatile matter and smoke which could not be burnt completely by secondary air.

6.2.4 Fluidized Bed Combustion

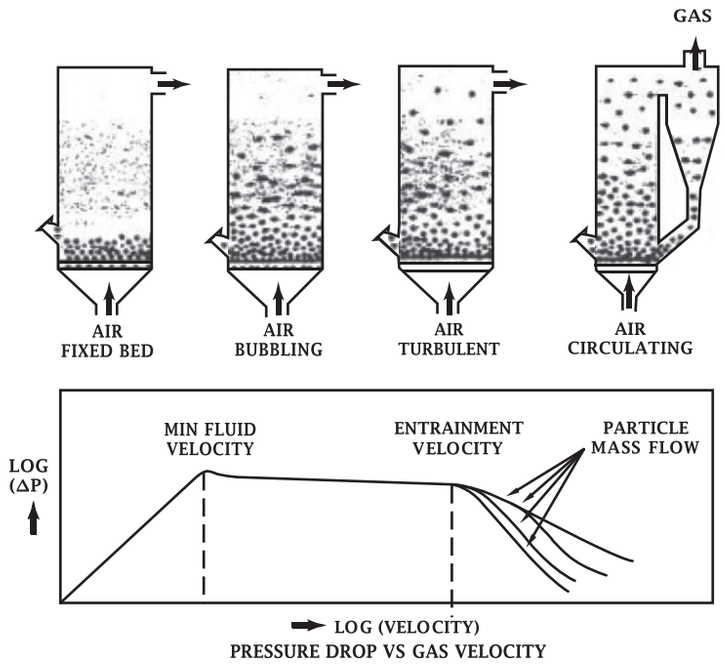
When an evenly distributed air or gas is passed upward through a finely divided bed of solid particles such as sand supported on a fine mesh, the particles are undisturbed at low velocity. As air velocity is

gradually increased, a stage is reached when the individual particles are suspended in the air stream – the bed is called "fluidized".

With further increase in air velocity, there is bubble formation, vigorous turbulence, rapid mixing and formation of dense defined bed surface. The bed of solid particles exhibits the properties of a boiling liquid and assumes the appearance of a fluid – "bubbling fluidized bed".

At higher velocities, bubbles disappear, and particles are blown out of the bed. Therefore, some amounts of particles have to be recirculated to maintain a stable system – "circulating fluidized bed".

This principle of fluidization is illustrated in Figure 6.1.



Fixing, bubbling and fast fluidized beds

As the velocity of a gas flowing through a bed of particles increases, a value is reaches when the bed fluidises and bubbles form as in a boiling liquid. At higher velocities the bubbles disappear; and the solids are rapidly blown out of the bed and must be recycled to maintain a stable system.

Figure 6.1 Principle of Fluidisation

Fluidization depends largely on the particle size and the air velocity. The mean solids velocity increases at a slower rate than does the gas velocity, as illustrated in Figure 6.2. The difference between the mean solid velocity and mean gas velocity is called as slip velocity. Maximum slip velocity between the solids and the gas is desirable for good heat transfer and intimate contact.

If sand particles in a fluidized state is heated to the ignition temperatures of coal and coal is injected continuously into the bed, the coal will burn rapidly and bed attains a uniform temperature. The fluidized bed combustion (FBC) takes place at about 840 ^{0}C to 950 ^{0}C. Since this temperature is much below the ash fusion temperature, melting of ash and associated problems are avoided.

The lower combustion temperature is achieved because of high coefficient of heat transfer due to rapid mixing in the fluidized bed and effective extraction of heat from the bed through in-bed heat transfer tubes and walls of the bed. The gas velocity is maintained between minimum fluidization velocity

and particle entrainment velocity. This ensures stable operation of the bed and avoids particle entrainment in the gas stream.

Combustion process requires the three "T"s that is Time, Temperature and Turbulence. In FBC, turbulence is promoted by fluidization. Improved mixing generates evenly distributed heat at lower temperature. Residence time is many times greater than conventional grate firing. Thus an FBC system releases heat more efficiently at lower temperatures.

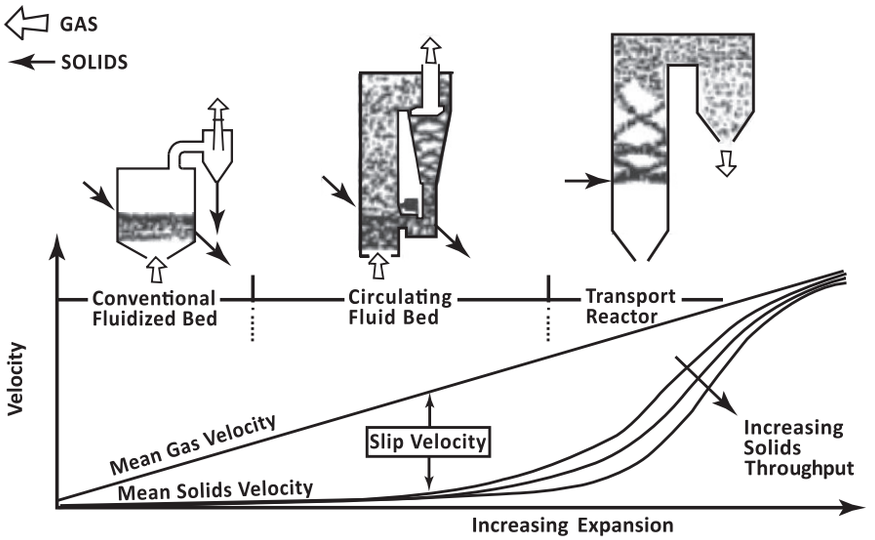


Figure 6.2 Relation between Gas Velocity and Solid Velocity

Since limestone is used as particle bed, control of sulfur dioxide and nitrogen oxide emissions in the combustion chamber is achieved without any additional control equipment. This is one of the major advantages over conventional boilers.

6.3 Types of Fluidized Bed Combustion Boilers

There are three basic types of fluidized bed combustion boilers:

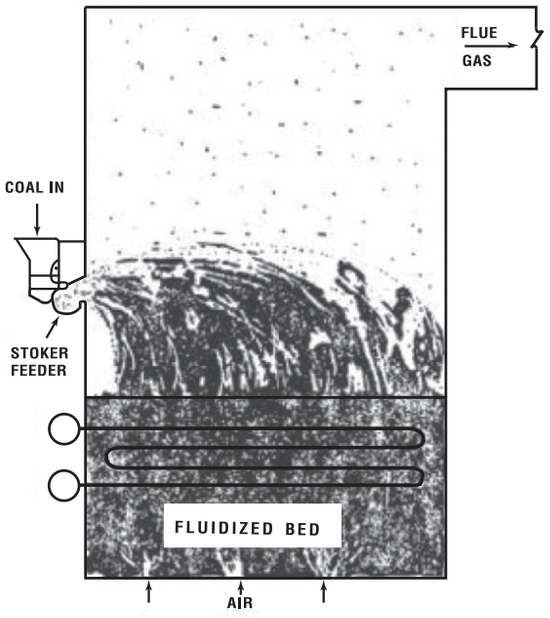
1. Atmospheric classic Fluidized Bed Combustion System (AFBC)
2. Atmospheric circulating (fast) Fluidized Bed Combustion system(CFBC)
3. Pressurized Fluidized Bed Combustion System (PFBC).

6.3.1 AFBC / Bubbling Bed

In AFBC, coal is crushed to a size of 1 – 10 mm depending on the rank of coal, type of fuel feed and fed into the combustion chamber. The atmospheric air, which acts as both the fluidization air and combustion air, is delivered at a pressure and flows through the bed after being preheated by the exhaust flue gases. The velocity of fluidizing air is in the range of 1.2 to 3.7 m /sec. The rate at which air is blown through the bed determines the amount of fuel that can be reacted.

Almost all AFBC/ bubbling bed boilers use in-bed evaporator tubes in the bed of limestone, sand and fuel for extracting the heat from the bed to maintain the bed temperature. The bed depth is usually 0.9

m to 1.5 m deep and the pressure drop averages about 1 inch of water per inch of bed depth. Very little material leaves the bubbling bed – only about 2 to 4 kg of solids is recycled per ton of fuel burned. Typical fluidized bed combustors of this type are shown in Figures 6.3 and 6.4.



Features of bubbling bed boiler

Fluidised bed boiler can operate at near atmospheric or elevated pressure and have these essential features:

- Distribution plate through which air is blown for fluidizing.
- Immersed steam-raising or water heating tubes which extract heat directly from the bed.
- Tubes above the bed which extract heat from hot combustion gas before it enters the flue duct.

Figure 6.3 Bubbling Bed Boiler-1

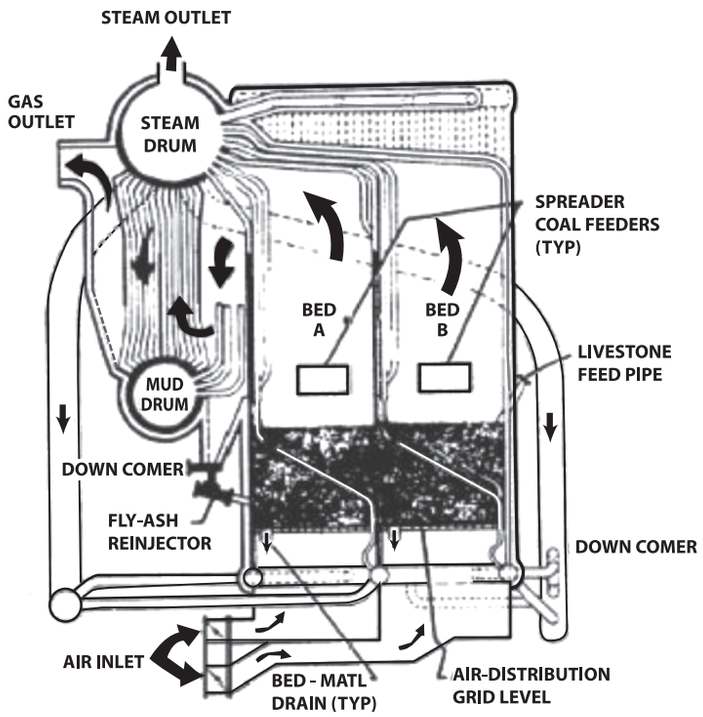


Figure 6.4 Bubbling Bed Boiler-2

The combustion gases pass over the super heater sections of the boiler, flow past the economizer, the dust collectors and the air pre heaters before being exhausted to atmosphere.

The main special feature of atmospheric fluidized bed combustion is the constraint imposed by the relatively narrow temperature range within which the bed must be operated. With coal, there is risk of clinker formation in the bed if the temperature exceeds 950 ^{0}C and loss of combustion efficiency if the temperature falls below 800 ^{0}C. For efficient sulphur retention, the temperature should be in the range of 800 ^{0}C to 850 ^{0}C.

General Arrangements of AFBC Boiler

AFBC boilers comprise of following systems:

i) Fuel feeding system
ii) Air Distributor
iii) Bed & In-bed heat transfer surface
iv) Ash handling system.

Many of these are common to all types of FBC boilers.

1. Fuel Feeding system

For feeding fuel, sorbents like limestone or dolomite, usually two methods are followed: under bed pneumatic feeding and over-bed feeding.

Under Bed Pneumatic Feeding

If the fuel is coal, it is crushed to 1-6 mm size and pneumatically transported from feed hopper to the combustor through a feed pipe piercing the distributor. Based on the capacity of the boiler, the number of feed points is increased, as it is necessary to distribute the fuel into the bed uniformly. In an AFBC boiler, coal feeder height may be increased above the bed level to reduce the primary air fan power consumption and also to reduce choking of coal feeding pipe by bed material during starting of the boilers.

Over-Bed Feeding

The crushed coal, 6-10 mm size is conveyed from coal bunker to a spreader by a screw conveyor. The spreader distributes the coal over the surface of the bed uniformly. This type of fuel feeding system accepts over size fuel also and eliminates transport lines, when compared to under-bed feeding system (Figure 6.5)

UNDER BED COAL NOZZLE **OVER BED COAL NOZZLE**

Figure 6.5 Under Bed and Over Bed Feeding

2. Air Distributor

The purpose of the distributor is to introduce the fluidizing air evenly through the bed cross section thereby keeping the solid particles in constant motion, and preventing the formation of defluidization zones within the bed. The distributor, which forms the furnace floor, is normally constructed from metal plate with a number of perforations in a definite geometric pattern. The perforations may be located in simple nozzles or nozzles with bubble caps, which serve to prevent solid particles from flowing back into the space below the distributor.

The distributor plate is protected from high temperature of the furnace by:

i) Refractory Lining
ii) A Static Layer of the Bed Material or
iii) Water Cooled Tubes.

3. Bed & In-Bed Heat Transfer Surface:

a) Bed

The bed material can be sand, ash, crushed refractory or limestone, with an average size of about 1 mm. Depending on the bed height these are of two types: shallow bed and deep bed.
At the same fluidizing velocity, the two ends fluidize differently, thus affecting the heat transfer to an immersed heat transfer surfaces. A shallow bed offers a lower bed resistance and hence a lower pressure drop and lower fan power consumption. In the case of deep bed, the pressure drop is more and this increases the effective gas velocity and also the fan power.

b) In-Bed Heat Transfer Surface

In a fluidized in-bed heat transfer process, it is necessary to transfer heat between the bed material and an immersed surface, which could be that of a tube bundle, or a coil. The heat exchanger orientation can be horizontal, vertical or inclined. From a pressure drop point of view, a horizontal bundle in a shallow bed is more attractive than a vertical bundle in a deep bed. Also, the heat transfer in the bed depends on number of parameters like (i) bed pressure (ii) bed temperature (iii) superficial gas velocity (iv) particle size (v) Heat exchanger design and (vi) gas distributor plate design.

4. Ash Handling System

a) Bottom ash removal

In the FBC boilers, the bottom ash constitutes roughly 30 - 40 % of the total ash, the rest being the fly ash. The bed ash is removed by continuous over flow to maintain bed height and also by intermittent flow from the bottom to remove over size particles, avoid accumulation and consequent defluidization. While firing high ash coal such as washery rejects, the bed ash overflow drain quantity is considerable so special care has to be taken.

b) Fly ash removal

The amount of fly ash to be handled in FBC boiler is relatively very high, when compared to conventional boilers. This is due to elutriation of particles at high velocities. Fly ash carried away by the flue gas is removed in number of stages; firstly in convection section, then from the bottom of air preheater/ economizer and finally a major portion is removed in dust collectors.
The types of dust collectors used are cyclone, bag filters, electrostatic precipitators (ESP's) or some combination of all of these. To increase the combustion efficiency, recycling of fly ash is practiced in some of the units.

c) Electrostatic Precipitator

Electrostatic precipitator is a method of dust collection that uses electrostatic force and consists of discharge wires and collecting plate. A high voltage is applied to the discharge wires to form an electric field between the wire and the collecting plate and it also ionizes the gas around the discharge wires to supply ions.

The flue gases containing dust flow between discharge wires and collecting plates, the dust particles in the gas are charged by the ions. Coulomb force caused by the electric field causes the charged particles to be collected on the collecting plate and the gas is purified. This is the principle of electrostatic precipitation, electrostatic precipitator (Figure 6.6) which is applied on the industrial scale. The dust collected on the collecting plates are removed by various methods such as rapping the collecting plate, washing off with water and removing from the hopper etc. ESPs can remove from 95% to 99% of fly ash from the flue gas.

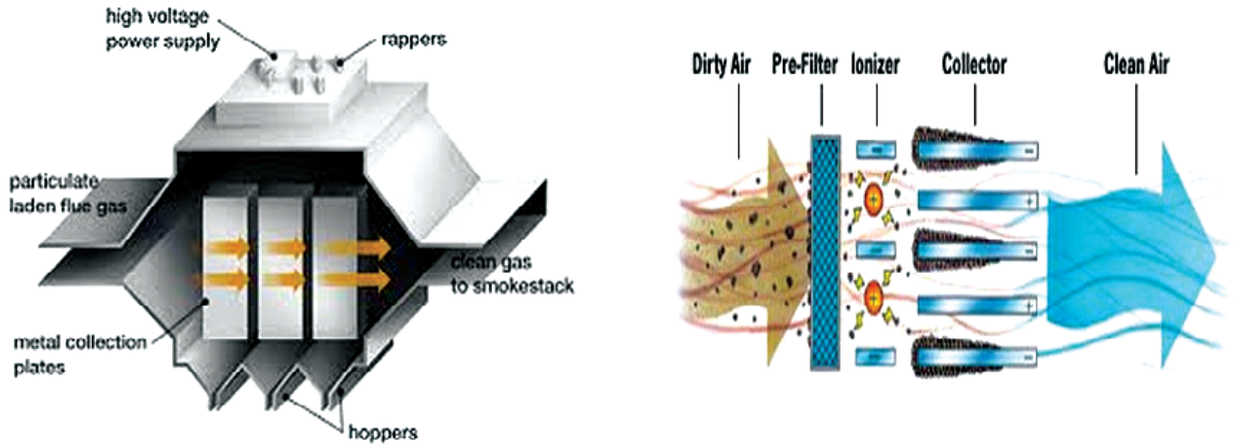


Figure 6.6 Electrostatic Precipitator

6.3.2 Circulating Fluidized Bed Combustion (CFBC)

Circulating Fluidized Bed Combustion (CFBC) technology has evolved from conventional bubbling bed combustion as a means to overcome some of the drawbacks associated with conventional bubbling bed combustion (see Figure 6.7).

This CFBC technology utilizes the fluidized bed principle in which crushed (6 –12 mm size) fuel and limestone are injected into the furnace or combustor. The particles are suspended in a stream of upwardly flowing air (60-70% of the total air), which enters the bottom of the furnace through air distribution nozzles. The fluidizing velocity in circulating beds ranges from 3.7 to 9 m/sec. The balance of combustion air is admitted above the bottom of the furnace as secondary air. The combustion takes place at 840-900 ^{0}C, and the fine particles (<450 microns) are elutriated out of the furnace with flue gas velocity of 4-6 m/s. The particles are then collected by the solids separators and circulated back into the furnace. Solid recycle is about 50 to 100 kg per kg of fuel burnt.

There are no steam generation tubes immersed in the bed. The circulating bed is designed to move a lot more solids out of the furnace area and to achieve most of the heat transfer outside the combustion zone - convection section, water walls, and at the exit of the riser. Some circulating bed units even have external heat exchanges.

The particles circulation provides efficient heat transfer to the furnace walls and longer residence time for carbon and limestone utilization. Similar to Pulverized Coal (PC) firing, the controlling parameters in the CFB combustion process are temperature, residence time and turbulence.

For large units, the taller furnace characteristics of CFBC boiler offers better space utilization, greater fuel particle and sorbent residence time for efficient combustion and SO_2 capture, and easier application of staged combustion techniques for NOx control than AFBC generators. CFBC boilers are said to achieve better calcium to sulphur utilization – 1.5 to 1 vs. 3.2 to 1 for the AFBC boilers, although the furnace temperatures are almost the same.

CFBC boilers are generally claimed to be more economical than AFBC boilers for industrial application requiring more than 75 – 100 T/hr. of steam

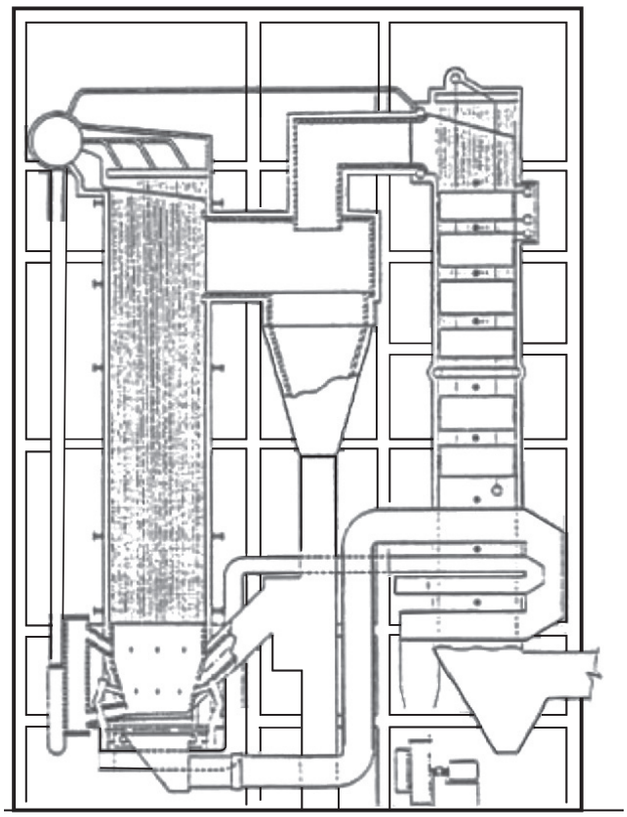
CFBC requires huge mechanical cyclones to capture and recycle the large amount of bed material, which requires a tall boiler.

A CFBC could be good choice if the following conditions are met.

- Capacity of boiler is large to medium
- Sulphur emission and NOx control is important
- The boiler is required to fire low-grade fuel or fuel with highly fluctuating fuel quality.

Major performance features of the circulating bed system are as follows:

a) It has a high processing capacity because of the high gas velocity through the system.
b) The temperature of about 870^{o}C is reasonably constant throughout the process because of the high turbulence and circulation of solids. The low combustion temperature also results in minimal NOx formation.
c) Sulfur present in the fuel is retained in the circulating solids in the form of calcium sulphate and removed in solid form. The use of limestone or dolomite sorbents allows a higher sulfur retention rate, and limestone requirements have been demonstrated to be substantially less than with bubbling bed combustor.
d) The combustion air is supplied at 1.5 to 2 psig rather than 3-5 psig as required by bubbling bed combustors.
e) It has high combustion efficiency.
f) It has a better turndown ratio than bubbling bed systems.
g) Erosion of the heat transfer surface in the combustion chamber is reduced, since the surface is parallel to the flow. In a bubbling bed system, the surface generally is perpendicular to the flow.



Circulating bed boiler

At high fluidizing gas velocities in which a fast recycling bed of fine material is superimposed on a bubbling bed of larger particles. The combustion temperature is controlled by rate of recycling of fine material. Hot fine material is separated from the flue gas by a cyclone and is partially cooled in a separate low velocity fluidized bed heat exchanger, where the heat is given up to the steam. The cooler fine material is then recycled to the dense bed.

Figure 6.7 Circulating Bed Boiler Design

6.3.3 Pressurized Fluid Bed Combustion

Pressurized Fluidized Bed Combustion (PFBC) is a variation of fluid bed technology that is meant for large-scale coal burning applications. In PFBC, the bed vessel is operated at pressure up to 16 ata (16 kg/cm^2).

The off-gas from the fluidized bed combustor drives the gas turbine. The steam turbine is driven by steam raised in tubes immersed in the fluidized bed. The condensate from the steam turbine is pre-heated using waste heat from gas turbine exhaust and is then taken as feed water for steam generation.

The PFBC system can be used for cogeneration or combined cycle power generation. By combining the gas and steam turbines in this way, electricity is generated more efficiently than in conventional system. The overall conversion efficiency is higher by 5% to 8%. (Refer Figure 6.8).

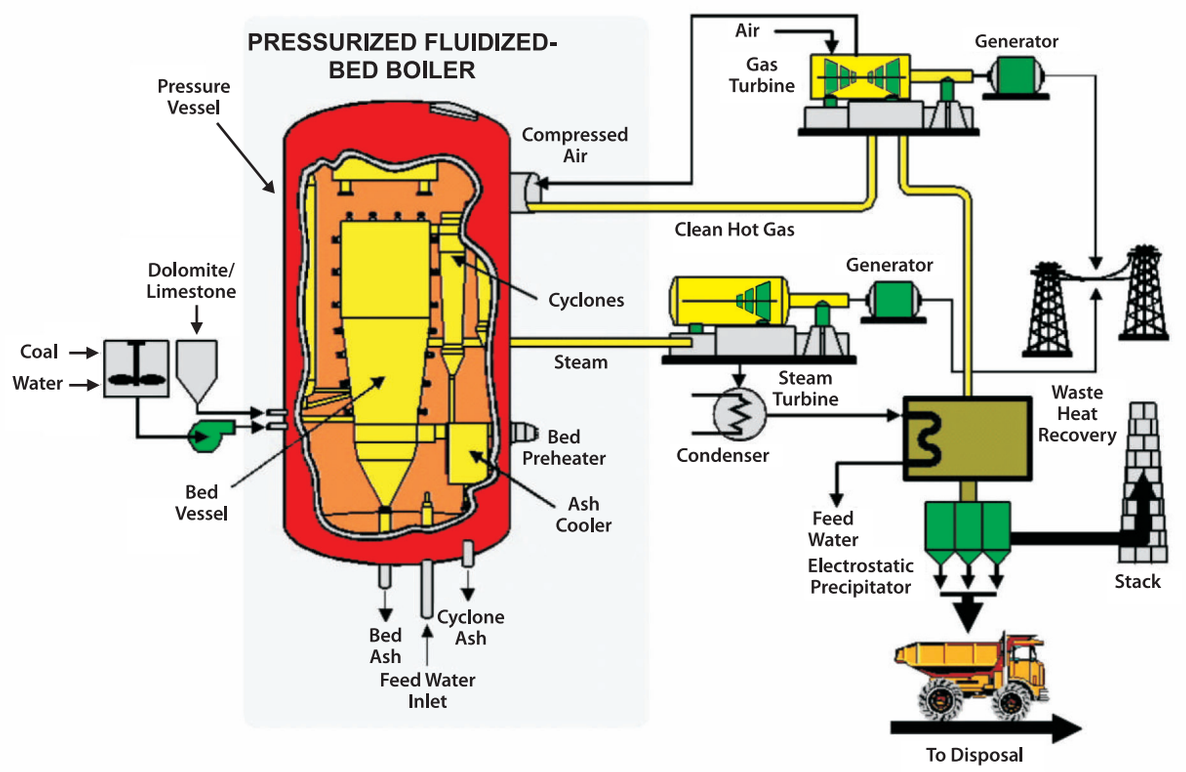


Figure 6.8 PFBC Boiler for Cogeneration

At elevated pressure, the potential reduction in boiler size is considerable due to increased amount of combustion in pressurized mode and high heat flux through in-bed tubes. A comparison of size of a typical 250 MW PFBC boiler versus conventional pulverized fuel-fired boiler is shown in the Figure 6.9.

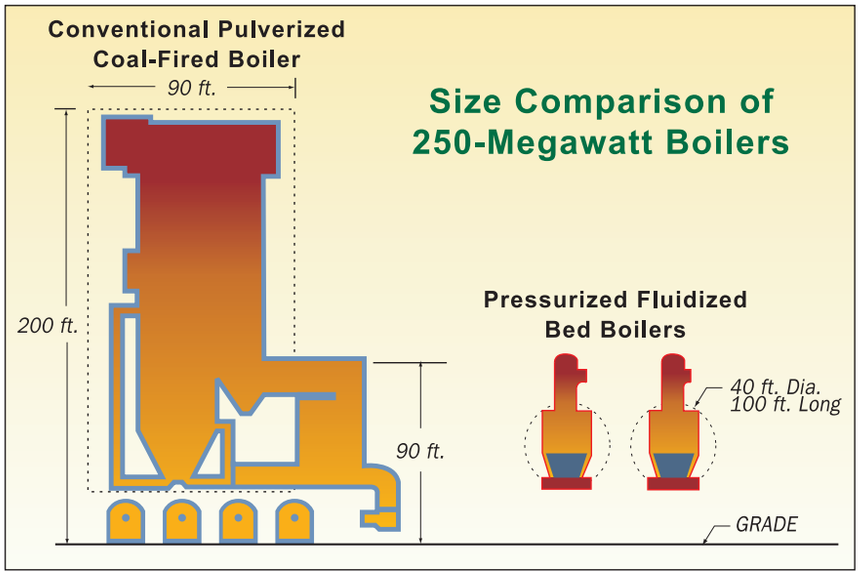


Figure 6.9 Comparison of PFBC Boiler versus Pulverized Fuel Boiler

6.4 Retrofitting of FBC Systems to Conventional Boilers

Retrofitting fluidized bed coal fired combustion systems to conventional boilers has been carried out successfully both in India and abroad.

The important aspects to be considered in retrofit projects are:

a) Water/steam circulation design
b) Furnace bottom-grate clearance
c) Type of particulate control device
d) Fan capacity
e) Availability of space.

Retrofitting of a fluidized bed combustor to a conventional stoker fired water tube boiler may involve:

a) The replacement of grate by a distributor plate with short stand pipes for admitting air from the wind box located underneath.
b) Installations of stand pipes to remove ash from the bed.
c) Provision of horizontal hairpin tubes in the bed with a pump for forced circulation from the boiler drum.
d) Modification of crusher to size the coal/limestone mixture for pneumatic under bed injection of the mixture.

It may be emphasized that conversion of a conventional coal fired system to a fluidized bed combustion system can be accomplished without effecting major changes, after making a cost-benefit analysis. Oil fired boilers can also be converted to coal fired fluidized bed combustion systems. However it has to be examined on a case-to-case basis.

6.5 Advantages of Fluidized Bed Combustion Boilers

1. High Efficiency

FBC boilers can burn fuel with a combustion efficiency of over 95% irrespective of ash content. FBC boilers can operate with overall efficiency of 84% (plus or minus 2%).

2. Reduction in Boiler Size

High heat transfer rate over a small heat transfer area immersed in the bed result in overall size reduction of the boiler.

3. Fuel Flexibility

FBC boilers can be operated efficiently with a variety of fuels. Even fuels like flotation slimes, washer rejects, agro waste can be burnt efficiently. These can be fed either independently or in combination with coal into the same furnace.

4. Ability to Burn Low Grade Fuel

FBC boilers would give the rated output even with inferior quality fuel. The boilers can fire coals with ash content as high as 62% and having calorific value as low as 2,500 kcal/kg. Even carbon content of only 1% by weight can sustain the fluidized bed combustion.

5. Ability to Burn Fines

Coal containing fines below 6 mm can be burnt efficiently in FBC boiler, which is very difficult to achieve in conventional firing system.

6. Pollution Control

SO_2 formation can be greatly minimized by addition of limestone or dolomite for high sulphur coals. 3% limestone is required for every 1% sulphur in the coal feed. Low combustion temperature eliminates NO_x formation.

7. Low Corrosion and Erosion

The corrosion and erosion effects are less due to lower combustion temperature, softness of ash and low particle velocity (of the order of 1 m/sec).

8. Easier Ash Removal – No Clinker Formation

Since the temperature of the furnace is in the range of 750 – 900 ^{0}C in FBC boilers, even coal of low ash fusion temperature can be burnt without clinker formation. Ash removal is easier as the ash flows like liquid from the combustion chamber. Hence less manpower is required for ash handling.

9. Less Excess Air – Higher CO_2 in Flue Gas

The CO_2 in the flue gases will be of the order of 14 – 15% at full load. Hence, the FBC boiler can operate at low excess air - only 20 – 25%.

10. Simple Operation, Quick Start-Up

High turbulence of the bed facilitates quick start up and shut down. Full automation of start up and operation using reliable equipment is possible.

11. Fast Response to Load Fluctuations

Inherent high thermal storage characteristics can easily absorb fluctuation in fuel feed rates. Response to changing load is comparable to that of oil fired boilers.

12. No Slagging in the Furnace-No Soot Blowing

In FBC boilers, volatilization of alkali components in ash does not take place and the ash is non-sticky. This means that there is no slagging or soot blowing.

13. Provisions of Automatic Coal and Ash Handling System

Automatic systems for coal and ash handling can be incorporated, making the plant easy to operate comparable to oil or gas fired installation.

14. Provision of Automatic Ignition System

Control systems using micro-processors and automatic ignition equipment give excellent control with minimum manual supervision.

15. High Reliability

The absence of moving parts in the combustion zone results in a high degree of reliability and low maintenance costs.

16. Reduced Maintenance

Routine overhauls are infrequent and high efficiency is maintained for long periods.

17. Quick Responses to Changing Demand

A fluidized bed combustor can respond to changing heat demands more easily than stoker fired systems. This makes it very suitable for applications such as thermal fluid heaters, which require rapid responses.

18. High Efficiency of Power Generation

By operating the fluidized bed at elevated pressure, it can be used to generate hot pressurized gases to power a gas turbine**.** This can be combined with a conventional steam turbine to improve the efficiency of electricity generation and give a potential fuel savings of at least 4%.

6.6. Application Considerations with Biomass FBC Boilers

Biomass is a clean source of energy and is gaining huge popularity for FBC application in Industry. Used judiciously, entrepreneurs can reduce the cost of steam generation and at the same time reduce greenhouse gas emissions. With biomass fed boilers, the following areas need attention:

- Uneven spreading of biomass fuel on boiler grate can lead to secondary combustion in the super-heater zone, resulting in overheating of super heater tubes and fluctuations in steam pressure.

- Frequent erosion of super-heater and economizer coils can occur, due to high silica content in the biomass, especially in rice husk.
- High extraneous matter in biomass (sand and mud) causes tube fouling and fluidized bed to be drained more frequently, with resultant heat loss.
- Carbon and dust coating of boiler tubes results in lowering of steam temperatures, especially during soot blowing.
- Presence of Pesticides (used during farming) adds to tube failure frequencies; mainly due to potassium constituents.
- Corrosive constituents in biomass adversely affect boiler internals, especially the super-heater tubes. Chloride content in certain types of biomass (like cotton stalk, 8–9%) can combine with sodium and potassium in high temperature regime to aggravate the corrosion process.
- Some boilers which use Red Gram husk/twigs as fuel pose corrosion problems at the cold end (i.e., secondary super-heater and economizer tubes), due to the sulfur content.
- The biomass fuel mix fed to the boiler, in quite a few cases, contains a combination of 6 to 7 biomass types. Each biomass has a separate air-to fuel ratio, and it is difficult to set a workable air–fuel ratio.
- High moisture content in the biomass causes frequent jamming of the fuel in feeders, leading to fluctuations in steam pressure and temperature.
- High moisture content in the biomass also leads to plugging and choking of closely spaced heating surfaces. This situation is further aggravated by the super-heater tube coil with very close spacing, often the result of a desire to achieve a compact design.
- Due to biomass fuel size variation, occurrence of unburnts in flue gases and bottom ash is high, resulting in lower efficiency and also variation in steam pressure and temperature.
- Absence of biomass feed rate measurement mechanism leaves little room for accurate assessment of heat rate/efficiency. Providing a weighing mechanism is difficult on account of different biomass fuel combinations being used, with different (and low) bulk densities.
- Degradation of biomass during storage in exposed ambient wet atmosphere leads to loss of heat value. Loss of material due to windage and carpet loss, coupled with loss of heat value on account of decay (inherent biomass characteristics), can cause an error in assessment of input fuel energy (as the input heat is customarily evaluated based on received biomass quantities and GCV).

QUESTIONS	
Objective Type Questions	
1.	Fluidized bed temperature exceeding 950 °C in a FBC boiler may result in a) sand heating b) ash fusion c) melting of lime stones d) ash carry over
2.	For even distribution of fluidized air in AFBC boilers, which one of the following is used? a) perforated metal distributor plate b) secondary air from sides c) compressed air through nozzles d) none of the above
3.	The difference between mean solid and mean gas velocity in FBC boiler is called a) fluidization factor b) slip velocity c) settling velocity d) none of the above
4.	In a CFBC boiler the capture and recycling of bed materials is accomplished by a) bag filter b) settling chamber c) cyclone d) scrubber system
5.	The efficiency of a typical FBC boiler would be around a) 90% b) 80% c) 60% d) 50%
6.	In FBC boiler the combustion is carried out at a temperature a) closer to steam temperature b) below ash fusion temperature of fuel used c) at adiabatic combustion temperature d) at and above ash fusion temperature
7.	Fuel bed temperature in a FBC boiler is about a) 550 °C b) 900°C c) 1400 °C d) 1700 °C
8.	Low combustion temperature in FBC boilers results in reduced formation of a) SO_x b) NO_x c) CO_2 d) O_2
9.	Electrical energy consumption for coal sizing will be maximum for a) stoker fired boiler b) AFBC boiler c) CFBC boiler d) pulverized coal boiler
10.	SO_x emissions in a FBC boiler fired with high sulfur coal are controlled by adding ____ to the bed a) Magnesia b) Limestone c) Sand d) Silica
Short Type Questions	
S-1	How SO_2 emissions are controlled in a FBC boiler and describe the method.
S-2	What are the important aspects to be considered for retrofitting a conventional boiler to FBC technology?
S-3	What is the principle of CFBC (circulating fluidized bed combustion) boiler?
S-4	Explain the difference between CFBC and AFBC
S-5	Explain the operating principle of PFBC Boiler.

Long Type Questions	
L-1	Explain the concept of a FBC boiler with a sketch. What are the major benefits of using a FBC boiler?
L-2	A chemical plant has an AFBC boiler with the following specifications: • Boiler capacity : 70 TPH • Boiler pressure : 60 kg/cm^2 • Steam temperature : 500^{o}C • Fuel fired : coal with 35% ash content • GCV of coal : 4000 kcal/kg • Theoretical air for combustion : 5.6 kg/kg of coal • Hydrogen in fuel : 4% • Specific heat of flue gas : 0.24 kcal/kg^{o}C • Specific heat of superheated water vapor in the flue gas : 0.45 kcal/kg^{o}C Operating parameters are given below • Flue gas exhaust temperature : 160^{o}C • Excess air : 30% • Feed water temperature : 105^{o}C • Radiation and other losses : 8% • Ambient temperature : 30^{o}C (i) Calculate the Boiler Efficiency using indirect method on GCV basis. (ii) If the feed water temperature is 105^{o}C and the steam is produced at 60 kg/ cm^2 & 500^{o}C, what is the hourly coal consumption? Total heat of steam at 60 kg/cm^2 & 500^{o}C is 817 kcal/kg.

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Chapter 7

Cogeneration

Learning Objectives

In this chapter you will learn about

- Need for cogeneration and advantages
- Basics of thermodynamic cycles
- Main types of cogeneration systems
- Heat to power ratio
- Steam and gas turbines
- Diesel engine systems
- Relative merits of various cogeneration systems
- Methods of turbine performance monitoring
- Trigeneration
- Micro turbine

7. COGENERATION

Definition, Need, Application, Advantages, Classification, Saving Potentials, Heat Balance, Steam Turbine Efficiency, Trigeneration, Micro turbine

7.1 Need for Cogeneration

Thermal power plants are a major source of electricity supply in India. The conventional method of power generation and supply to the customer is wasteful in the sense that only about a third of the primary energy fed into the power plant is actually made available to the user in the form of electricity (Figure 7.1). In conventional power plant, efficiency is only 35% and remaining 65% of energy is lost. The major source of loss in the conversion process is the heat rejected to the surrounding water or air due to the inherent constraints of the different thermodynamic cycles employed in power generation. Also further losses of around 10-15% are associated with the transmission and distribution of electricity in the electrical grid.

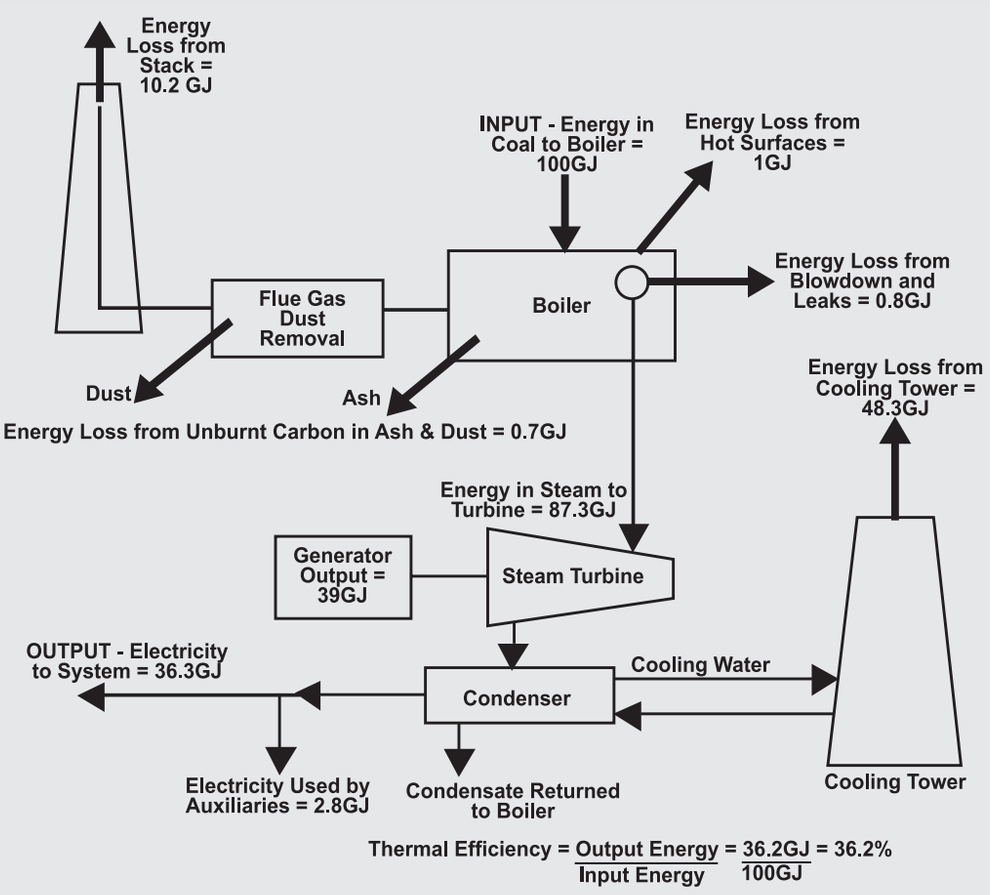


Figure 7.1 BALANCE IN A TYPICAL COAL FIRED POWER STATION For an Input Energy of 100 Giga Joules (GJ)

7.2 Principle of Cogeneration

Cogeneration or Combined Heat and Power (CHP) is defined as the sequential generation of two different forms of useful energy from a single primary energy source, typically mechanical energy and thermal energy. Mechanical energy may be used either to drive an alternator for producing electricity, or rotating equipment such as motor, compressor, pump or fan for delivering various services. Thermal energy can be used either for direct process applications or for indirectly producing steam, hot water, hot air for dryer or chilled water for process cooling.

Cogeneration provides a wide range of technologies for application in various domains of economic activities. The overall efficiency of energy use in cogeneration mode can be up to 85 per cent and above in some cases.

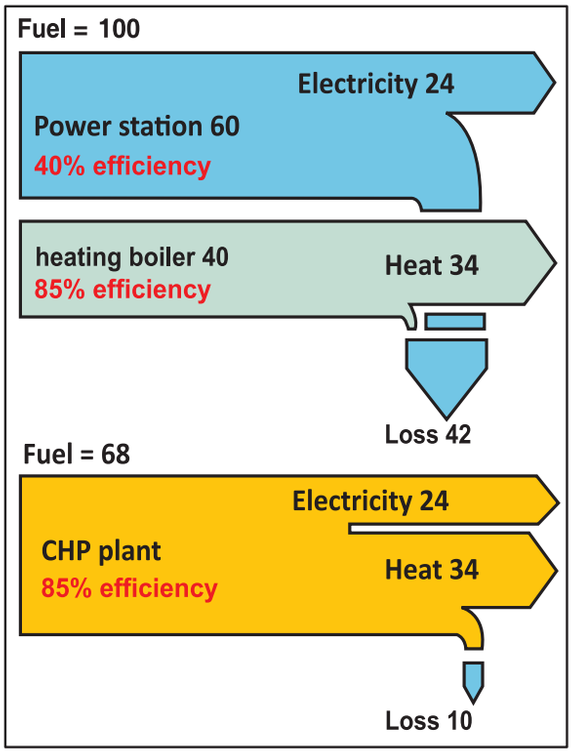


Figure 7.2 Cogeneration Advantage

For example in the scheme shown in Figure 7.2, an industry requires 24 units of electrical energy and 34 units of heat energy. Through separate heat and power route the primary energy input in power plant will be 60 units (24/0.40). If a separate boiler is used for steam generation then the fuel input to boiler will be 40 units (34/0.85). If the plant had cogeneration then the fuel input will be only 68 units (24+34)/0.85 to meet both electrical and thermal energy requirements. It can be observed that the losses, which were 42 units in the case of, separate heat and power has reduced to 10 units in cogeneration mode.

Along with the saving of fossil fuels, cogeneration also allows to reduce the emission of greenhouse gases (particularly CO_2 emission). The production of electricity being on-site, the burden on the utility network is reduced and the transmission line losses eliminated.

Cogeneration makes sense from both macro and micro perspectives. At the macro level, it allows a part of the financial burden of the national power utility to be shared by the private sector; in addition, indigenous energy sources are conserved. At the micro level, the overall energy bill of the users can be reduced, particularly when there is a simultaneous need for both power and heat at the site, and a rational energy tariff is practiced in the country.

7.3 Technical Options for Cogeneration

Cogeneration technologies that have been widely commercialized include extraction/back pressure steam turbines, gas turbine with heat recovery boiler (with or without bottoming steam turbine) and reciprocating engines with heat recovery boiler.

7.3.1 Steam turbine cogeneration systems

The two types of steam turbines most widely used are the backpressure and the extraction-condensing types (see Figure 7.3). The choice between backpressure turbine and extraction-condensing turbine depends mainly on the quantities of power and heat, quality of heat, and economic factors. The extraction points of steam from the turbine could be more than one, depending on the temperature levels of heat required by the processes.

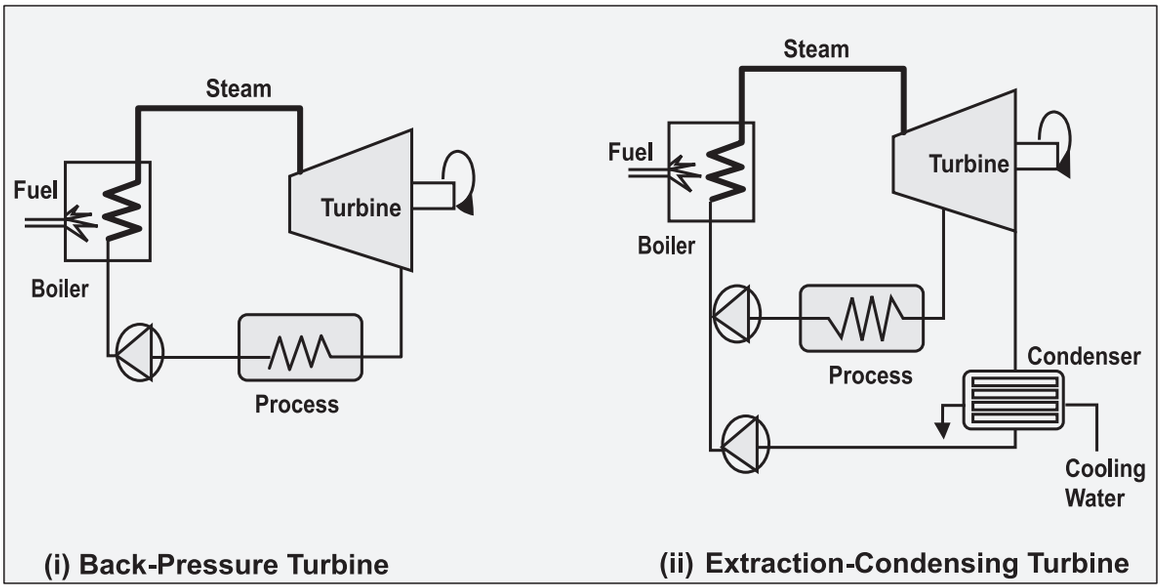


Figure 7.3 Schematic Diagrams of Steam Turbine Co-generation Systems

Another variation of the steam turbine topping cycle cogeneration system is the extraction-back pressure turbine that can be employed where the end-user needs thermal energy at two different temperature levels. The full-condensing steam turbines are usually incorporated at sites where heat rejected from the process is used to generate power.

The specific advantage of using steam turbines in comparison with the other prime movers is the option for using a wide variety of conventional as well as alternative fuels such as coal, natural gas, fuel oil and biomass. The power generation efficiency of the cycle may be sacrificed to some extent in order to optimize heat supply. In backpressure cogeneration plants, there is no need for large cooling towers. Steam turbines are mostly used where the demand for electricity is greater than one MW up to a few hundreds of MW. Due to the system inertia, their operation is not suitable for sites with intermittent energy demand.

7.3.2 Gas turbine cogeneration systems

Gas turbine cogeneration systems can produce all or a part of the energy requirement of the site, and the energy released at high temperature in the exhaust stack can be recovered for various heating and cooling applications (see Figure 7.4). Though natural gas is most commonly used, other fuels such as light fuel oil or diesel can also be employed. The typical range of gas turbines varies from a fraction of a MW to around 100 MW.

Gas turbine cogeneration has probably experienced the most rapid development in the recent years due to the greater availability of natural gas, rapid progress in the technology, significant reduction in installation costs, and better environmental performance. Furthermore, the gestation period for developing a project is shorter and the equipment can be delivered in a modular manner. Gas turbine has a short start-up time and provides the flexibility of intermittent operation. Though it has a low heat to power conversion efficiency, more heat can be recovered at higher temperatures. If the heat output is less than that required by the user, it is possible to have supplementary natural gas firing by mixing additional fuel to the oxygen-rich exhaust gas to boost the thermal output more efficiently.

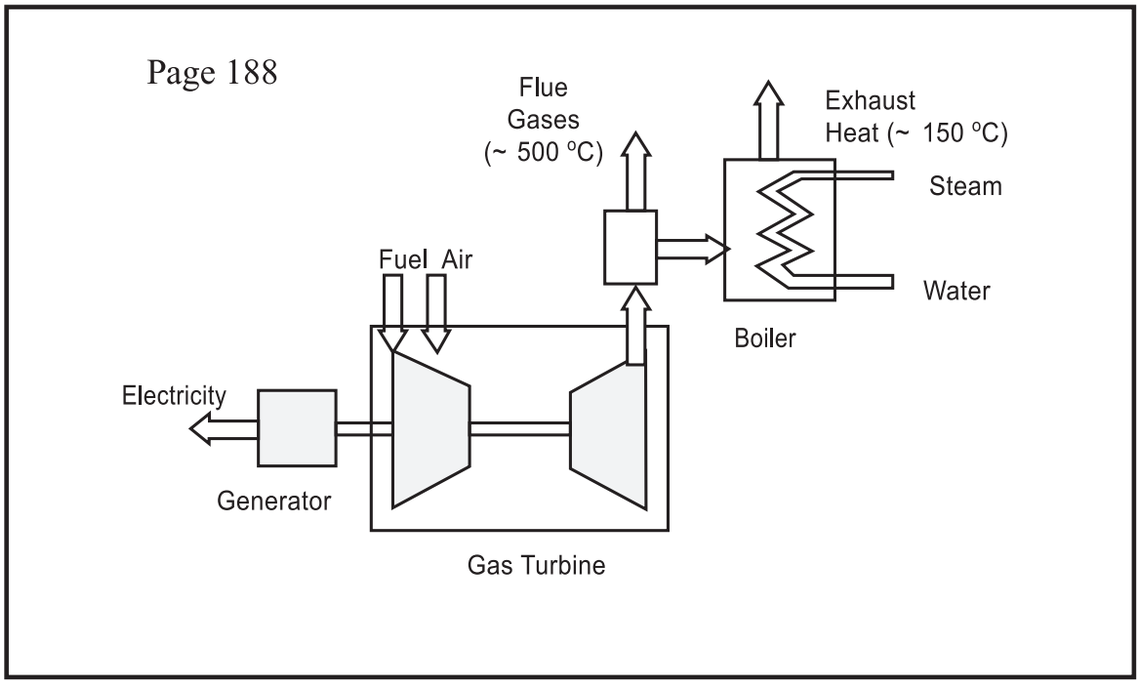


Figure 7.4 Schematic Diagram of Gas Turbine Cogeneration

On the other hand, if more power is required at the site, it is possible to adopt a combined cycle that is a combination of gas turbine and steam turbine cogeneration. Steam generated from the exhaust gas of the gas turbine is passed through a backpressure or extraction-condensing steam turbine to generate additional power. The exhaust or the extracted steam from the steam turbine provides the required thermal energy.

7.3.3 Reciprocating engine cogeneration systems

Also known as internal combustion (I. C.) engines, these cogeneration systems have high power generation efficiencies in comparison with other prime movers. There are two sources of heat for recovery: exhaust gas at high temperature and engine jacket cooling water system at low temperature (see Figure 7.5). As heat recovery can be quite efficient for smaller systems, these systems are more popular with smaller energy consuming facilities, particularly those having a greater need for electricity than thermal energy and where the quality of heat required is not high, e.g. low pressure steam or hot water.

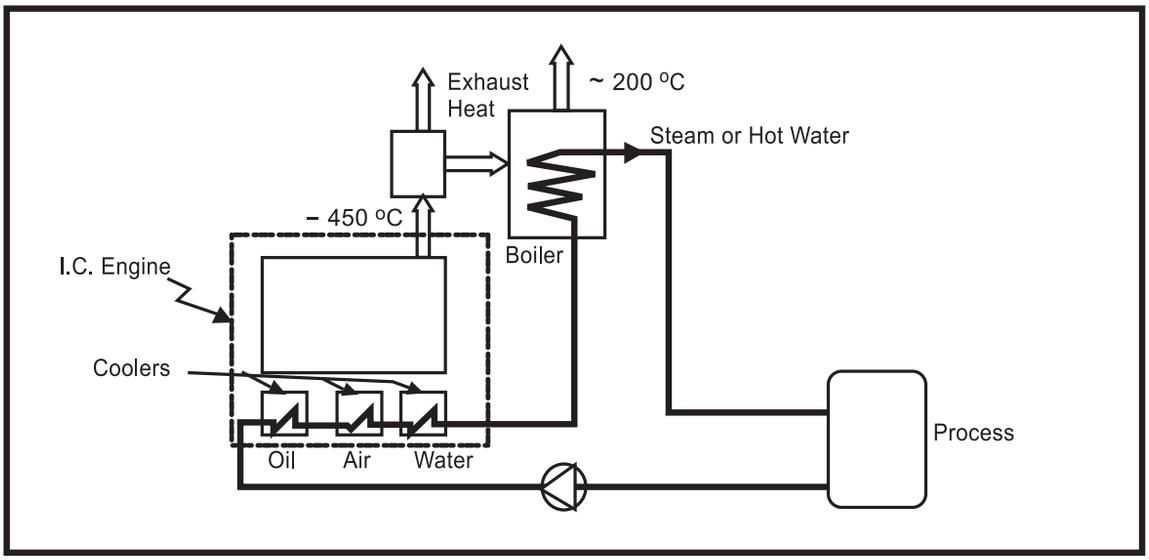


Figure 7.5 Schematic diagram of reciprocating engine cogeneration

Though diesel has been the most common fuel in the past, the prime movers can also operate with heavy fuel oil or natural gas. These machines are ideal for intermittent operation and their performance is not as sensitive to the changes in ambient temperatures as the gas turbines. Though the initial investment on these machines is low, their operating and maintenance costs are high due to high wear and tear.

7.4 Classification of Cogeneration Systems

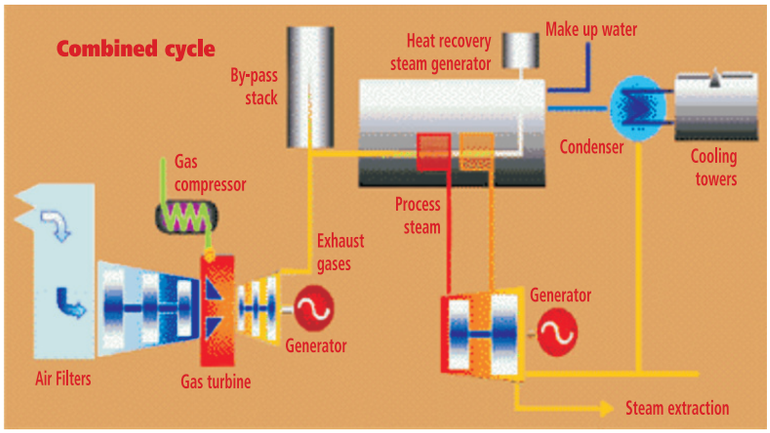
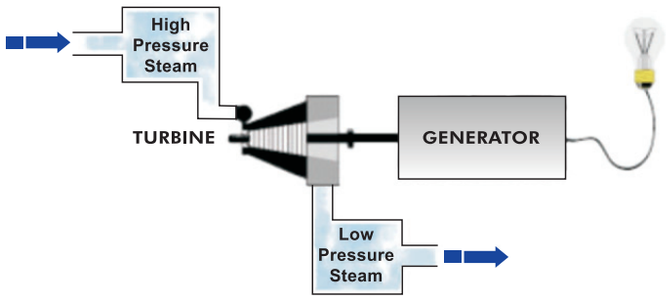
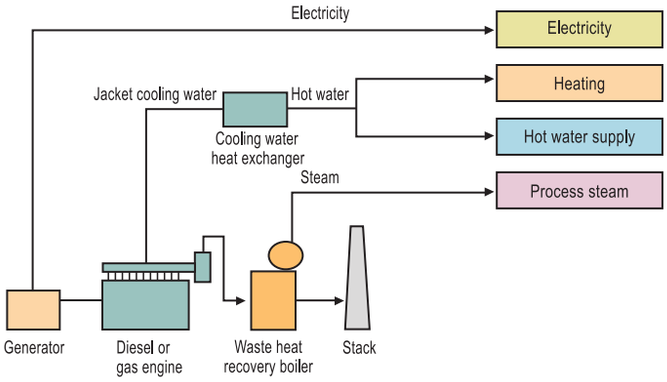
Cogeneration systems are normally classified according to the sequence of energy use and the operating schemes adopted.

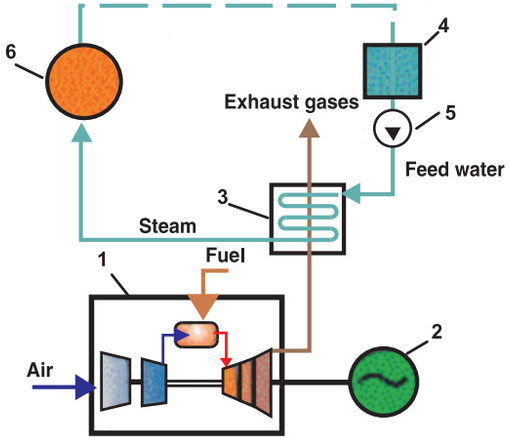
A cogeneration system can be classified as either a topping or a bottoming cycle on the basis of the sequence of energy use. In a topping cycle, the fuel supplied is used to first produce power and then thermal energy, which is the by-product of the cycle and is used to satisfy process heat or other thermal requirements. Topping cycle cogeneration is widely used and is the most popular method of cogeneration.

Topping Cycle

The four types of topping cycle cogeneration systems are briefly explained in Table 7.1.

Table 7.1 Types of Topping Cycles

A gas turbine or diesel engine producing electrical or mechanical power followed by a heat recovery boiler to create steam to drive a secondary steam turbine. This is called a combined-cycle topping system.	
The second type of system burns fuel (any type) to produce high-pressure steam that then passes through a steam turbine to produce power with the exhaust provides low-pressure process steam. This is a steam-turbine topping system.	
A third type employs heat recovery from an engine exhaust and/or jacket cooling system flowing to a heat recovery boiler, where it is converted to process steam / hot water for further use.	

The fourth type is a gas-turbine topping system. A natural gas turbine drives a generator. The exhaust gas goes to a heat recovery boiler that makes process steam and process heat	

Bottoming Cycle

In a bottoming cycle, the primary fuel produces high temperature thermal energy and the heat rejected from the process is used to generate power through a recovery boiler and a turbine generator. Bottoming cycles are suitable for manufacturing processes that require heat at high temperature in furnaces and kilns, and reject heat at significantly high temperatures. Typical areas of application include cement, steel, ceramic, gas and petrochemical industries. Bottoming cycle plants are much less common than topping cycle plants. The Figure 7.6 illustrates the bottoming cycle where fuel is burnt in a furnace to produce synthetic rutile. The waste gases coming out of the furnace is utilized in a boiler to generate steam, which drives the turbine to produce electricity.

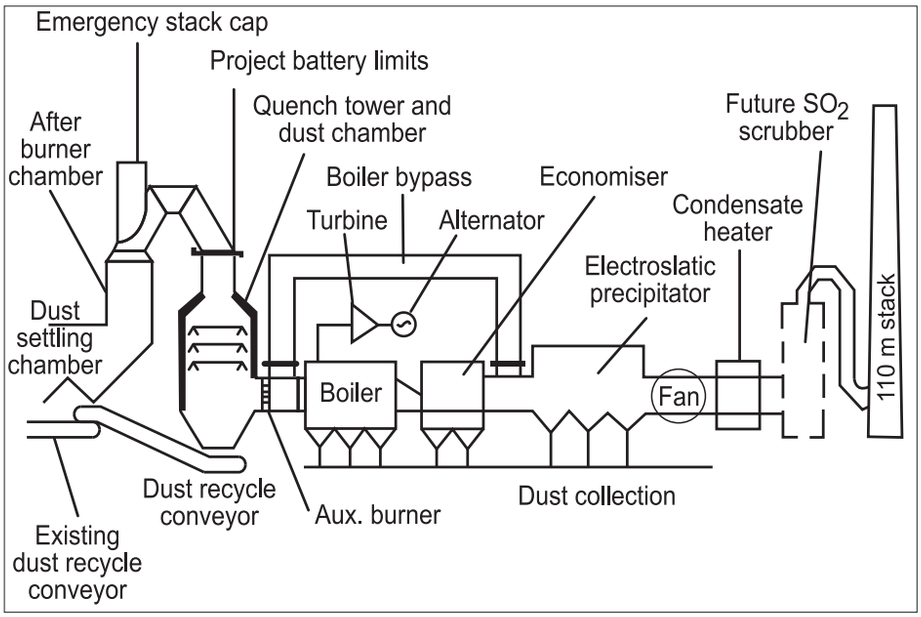


Figure 7.6 Bottoming Cycle

7.5 Factors Influencing Cogeneration Choice

The selection and operating scheme of a cogeneration system is very much site-specific and depends on several factors, as described below:

7.5.1 Base electrical load matching

In this configuration, the cogeneration plant is sized to meet the minimum electricity demand of the site based on the historical demand curve. The rest of the needed power is purchased from the utility grid. The thermal energy requirement of the site could be met by the cogeneration system alone or by additional boilers. If the thermal energy generated with the base electrical load exceeds the plant's demand and if the situation permits, excess thermal energy can be exported to neighbouring customers.

7.5.2 Base thermal load matching

Here, the cogeneration system is sized to supply the minimum thermal energy requirement of the site. Stand-by boilers or burners are operated during periods when the demand for heat is higher. The prime mover installed operates at full load at all times. If the electricity demand of the site exceeds that which can be provided by the prime mover, then the remaining amount can be purchased from the grid. Likewise, if local laws permit, the excess electricity can be sold to the power utility.

7.5.3 Electrical load matching

In this operating scheme, the facility is totally independent of the power utility grid. All the power requirements of the site, including the reserves needed during scheduled and unscheduled maintenance, are to be taken into account while sizing the system. This is also referred to as a "stand-alone" system. If the thermal energy demand of the site is higher than that generated by the cogeneration system, auxiliary boilers are used. On the other hand, when the thermal energy demand is low, some thermal energy is wasted. If there is a possibility, excess thermal energy can be exported to neighbouring facilities.

7.5.4 Thermal load matching

The cogeneration system is designed to meet the thermal energy requirement of the site at any time. The prime movers are operated following the thermal demand. During the period when the electricity demand exceeds the generation capacity, the deficit can be compensated by power purchased from the grid. Similarly, if the local legislation permits, electricity produced in excess at any time may be sold to the utility.

7.6 Important Technical Parameters for Cogeneration

While selecting cogeneration systems, one should consider some important technical parameters that assist in defining the type and operating scheme of different alternative cogeneration systems to be selected.

7.6.1 Heat-to-power ratio and Energy Utilisation factor

Heat-to-power ratio is one of the most important technical parameters influencing the selection of the type of cogeneration system. The heat-to-power ratio of a facility should match with the characteristics of the cogeneration system to be installed.

It is defined as the ratio of thermal energy to electricity required by the energy consuming facility. Though it can be expressed in different units such as Btu/kWh, kcal/kWh, lb./hr/kW, etc., here it is presented on the basis of the same energy unit (kW).

Basic heat-to-power ratios of the different cogeneration systems are shown in Table 7.2 along with some technical parameters. The steam turbine cogeneration system can offer a large range of heat-to-power ratios.

Table 7.2 Heat-to-Power Ratios and Other Parameters of Cogeneration Systems

Cogeneration System	**Heat-to-power ratio** (kW_{th} / kW_e)	**Power output** (as per cent of fuel input)	**Overall efficiency** (%)
Back-pressure steam turbine	4.0-14.3	14-28	84-92
Extraction-condensing steam turbine	2.0-10.0	22-40	60-80
Gas turbine	1.3-2.0	24-35	70-85
Combined cycle	1.0-1.7	34-40	69-83
Reciprocating engine	1.1-2.5	33-53	75-85

Cogeneration uses a single process to generate both electricity and usable heat or cooling. The proportions of heat and power needed (heat: power ratio) vary from site to site, so the type of plant must be selected carefully and appropriate operating schemes must be established to match demands as closely as possible. The plant may therefore be set up to supply part or all of the site heat and electricity loads, or an excess of either may be exported if a suitable customer is available. The following Table 7.3 shows typical heat:power ratios for certain energy intensive industries.

Table 7.3: Typical Heat: Power Ratios for Certain Energy Intensive Industries			
Industry	**Minimum**	**Maximum**	**Average**
Breweries	1.1	4.5	3.1
Pharmaceuticals	1.5	2.5	2.0
Fertilizer	0.8	3.0	2.0
Food	0.8	2.5	1.2
Paper	1.5	2.5	1.9

Cogeneration is likely to be most attractive under the following circumstances:

(a) The demand for both steam and power is balanced i.e. consistent with the range of steam: power output ratios that can be obtained from a suitable cogeneration plant.
(b) A single plant or group of plants has sufficient demand for steam and power to permit economies of scale to be achieved.
(c) Peaks and troughs in demand can be managed or, in the case of electricity, adequate backup supplies can be obtained from the utility company.

The ratio of heat to power required by a site may vary during different times of the day and seasons of the year. Importing power from the grid can make up a shortfall in electrical output from the cogeneration unit and firing standby boilers can satisfy additional heat demand. Many large cogeneration units utilize supplementary or boost firing of the exhaust gases in order to modify the heat: power ratio of the system to match site loads.

Energy Utilisation Factor, EUF is the percentage of useful output (i.e.) a combination of electrical output and thermal output to fuel heat input. The formula for EUF is given below.

$$Energy\ Utilisation\ Factor,\ EUF = \frac{Electrical\ output, Q_e + Thermal\ ouput, Q_{th}}{Fuel\ heat\ input, Q_f}$$

The electrical output is the useful power generation from the turbine and the thermal output is the useful extraction from turbine for process heating. The fuel heat input is the fuel flow rate multiplied by its calorific value.

7.6.2 Quality of thermal energy needed

The quality of thermal energy required (temperature and pressure) also determines the type of cogeneration system. For a sugar mill needing thermal energy at about 120°C, a topping cycle cogeneration system can meet the heat demand. On the other hand, for a cement plant requiring thermal energy at about 1450°C, a bottoming cycle cogeneration system can meet both high quality thermal energy and electricity demands of the plant.

7.6.3 Load patterns

The heat and power demand patterns of the user affect the selection (type and size) of the cogeneration system. For instance, the load patterns of two energy consuming facilities shown in Figure 7.7 would lead to two different sizes, possibly types also, of cogeneration systems.

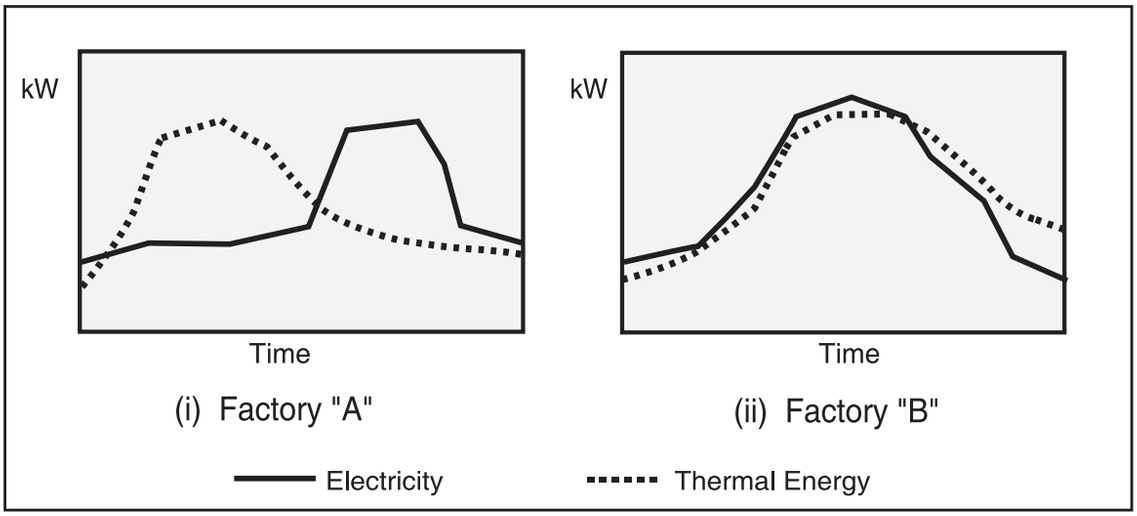


Figure 7.7 Different Heat and Power Demand Patterns in Two Factories

7.6.4 Fuels available

Depending on the availability of fuels, some potential cogeneration systems may have to be rejected. The availability of cheap fuels or waste products that can be used as fuels at a site is one of the major factors in the technical consideration because it determines the competitiveness of the cogeneration system.

A rice mill needs mechanical power for milling and heat for paddy drying. If a cogeneration system were considered, the steam turbine system would be the first priority because it can use the rice husk as the fuel, which is available as waste product from the mill.

7.6.5 System reliability

Some energy consuming facilities require very reliable power and/or heat; for instance, a pulp and paper industry cannot operate with a prolonged unavailability of process steam. In such instances, the cogeneration system to be installed must be modular, i.e. it should consist of more than one unit so that shut down of a specific unit cannot seriously affect the energy supply.

7.6.6 Grid dependent system versus independent system

A grid-dependent system has access to the grid to buy or sell electricity. The grid-independent system is also known as a "stand-alone" system that meets all the energy demands of the site. It is obvious that for the same energy consuming facility, the technical configuration of the cogeneration system designed as a grid dependent system would be different from that of a stand-alone system.

7.6.7 Retrofit versus new installation

If the cogeneration system is installed as a retrofit, the system must be designed so that the existing energy conversion systems, such as boilers, can still be used. In such a circumstance, the options for cogeneration system would depend on whether the system is a retrofit or a new installation.

7.6.8 Electricity buy-back

The technical consideration of cogeneration system must take into account whether the local regulations permit electric utilities to buy electricity from the cogenerators or not. The size and type of cogeneration system could be significantly different if one were to allow the export of electricity to the grid.

7.6.9 Local environmental regulation

The local environmental regulations can limit the choice of fuels to be used for the proposed cogeneration systems. If the local environmental regulations are stringent, some available fuels cannot be considered because of the high treatment cost of the polluted exhaust gas and in some cases, the fuel itself.

7.7 Prime Movers for Cogeneration

7.7.1 Steam Turbine

Steam turbines (Figure 7.8) are the most commonly employed prime movers for cogeneration applications in the steam turbine; the incoming high pressure steam is expanded to a lower pressure level, converting the thermal energy of high pressure steam to kinetic energy through nozzles and then to mechanical power through rotating blades.

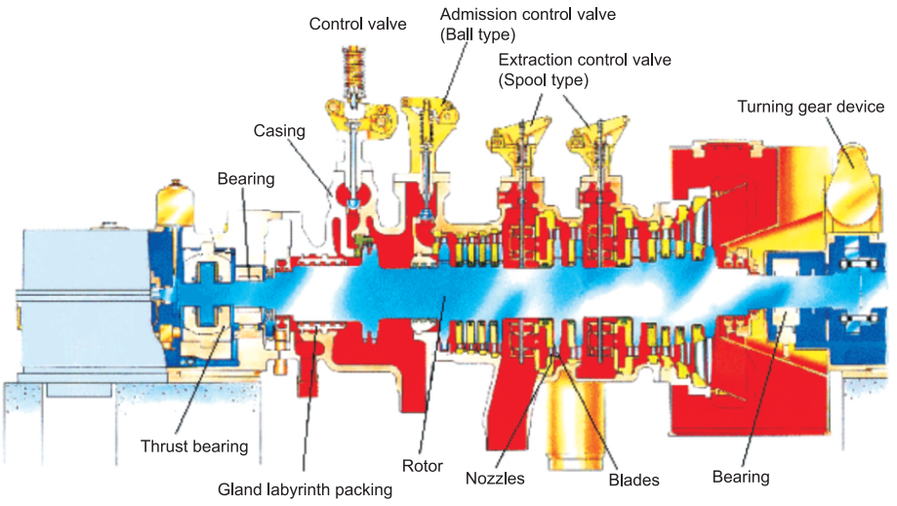


Figure 7.8 Steam Turbine

Back Pressure turbine: In this type steam enters the turbine chamber at High Pressure and expands to Low or Medium Pressure. Enthalpy difference is used for generating power / work.

Depending on the pressure (or temperature) levels at which process steam is required, backpressure steam turbines can have different configurations as shown in Figure 7.9.

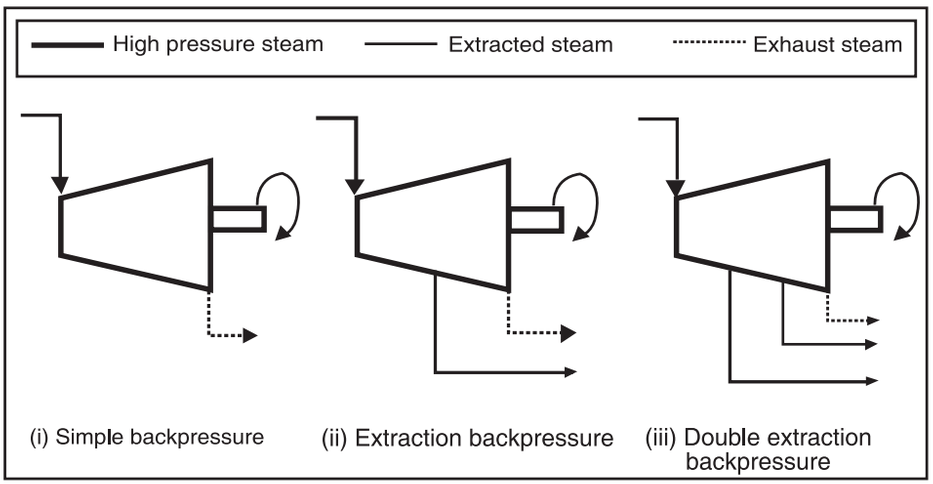


Figure 7.9 Different Configurations for Back Pressure Steam Turbines

In extraction and double extraction backpressure turbines, some amount of steam is extracted from the turbine after being expanded to a certain pressure level. The extracted steam meets the heat demands at pressure levels higher than the exhaust pressure of the steam turbine.

The efficiency of a backpressure steam turbine cogeneration system is the highest. In cases where 100 per cent backpressure exhaust steam is used, the only inefficiencies are gear drive and electric generator losses, and the inefficiency of steam generation. Therefore, with an efficient boiler, the overall thermal efficiency of the system could reach as much as 90 per cent.

Extraction Condensing turbine: In this type, steam entering at High / Medium Pressure is extracted at an intermediate pressure in the turbine for process use while the remaining steam continues to expand and condenses in a surface condenser and work is done till it reaches the Condensing pressure.(vacuum).

In Extraction cum Condensing steam turbine as shown in Figure 7.10, high Pressure steam enters the turbine and passes out from the turbine chamber in stages. In a two stage extraction cum condensing turbine MP steam and LP steam pass out to meet the process needs. Balance quantity condenses in the surface condenser. The Energy difference is used for generating Power. This configuration meets the heat-power requirement of the process.

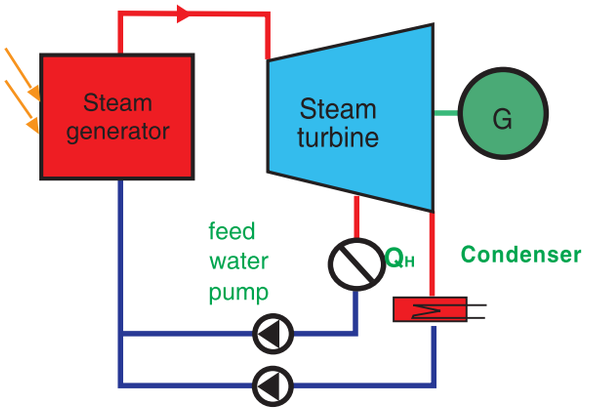


Figure 7.10 Extraction Condensing

The extraction condensing turbines have higher power to heat ratio in comparison with backpressure turbines. Although condensing systems need more auxiliary equipment such as the condenser and cooling towers, better matching of electrical power and heat demand can be obtained where electricity demand is much higher than the steam demand and the load patterns are highly fluctuating.

The overall thermal efficiency of an extraction condensing turbine cogeneration system is lower than that of back pressure turbine system, basically because the exhaust heat cannot be utilized (it is normally lost in the cooling water circuit). However, extraction condensing cogeneration systems have higher electricity generation efficiencies.

7.7.2 Gas Turbine

The fuel is burnt in a pressurized combustion chamber using combustion air supplied by a compressor that is integral with the gas turbine. In conventional Gas turbine (Figure 7.11), gases enter the turbine at a temperature range of 900 to 1000 ^{0}C and leave at 400 to 500 ^{0}C. The very hot pressurized gases are used to turn a series of turbine blades, and the shaft on which they are mounted, to produce mechanical energy. Residual energy in the form of a high flow of hot exhaust gases can be used to meet, wholly or partly, the thermal (steam) demand of the site. Waste gases are exhausted from the turbine at 450 ^{0}C to 550 ^{0}C, making the gas turbine particularly suitable for high-grade heat supply.

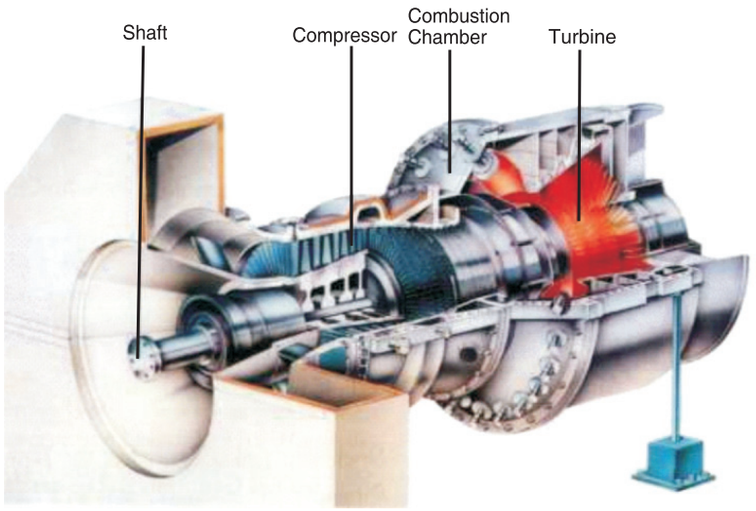


Figure 7.11 Gas Turbine

The available mechanical energy can be applied in the following ways:

- To produce electricity with a generator (most applications);
- To drive pumps, compressors, blowers, etc.

A gas turbine operates under exacting conditions of high speed and high temperature. The hot gases supplied to it must therefore be clean (i.e. free of particulates which would erode the blades) and must contain not more than minimal amounts of contaminants, which would cause corrosion under operating conditions. High-premium fuels are therefore most often used, particularly natural gas. Distillate oils such as gas oil are also suitable, and sets capable of using both are often installed to take advantage of cheaper interruptible gas tariffs. LPGs and Naphtha are also suitable, LPG being a possible fuel in either gaseous or liquid form.

Gas Turbine Efficiency

Turbine Efficiency is the ratio of actual work output of the turbine to the net input energy supplied in the form of fuel. For stand-alone Gas Turbines, without any heat recovery system the efficiency will be as low as 35 to 40%. This is attributed to the blade efficiency of the rotor, leakage through clearance spaces, friction, irreversible turbulence etc.

Since Exhaust gas from the Gas Turbine is high, it is possible to recover energy from the hot gas by a Heat Recovery Steam Generator and use the steam for process.

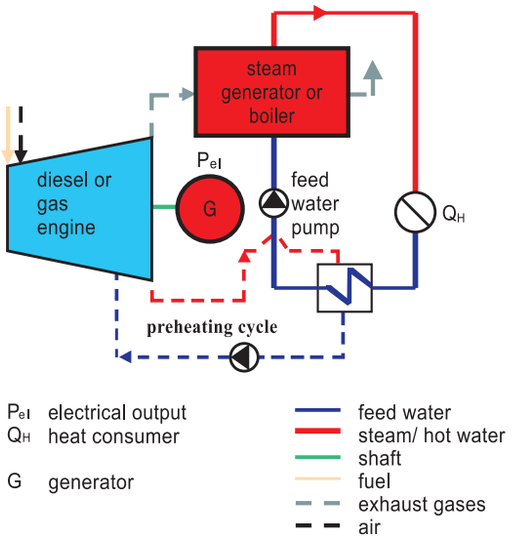
Net Turbine Efficiency

Above efficiency figures did not include the energy consumed by air compressors, fuel pump and other auxiliaries. Air compressor alone consumes about 50 to 60 % of energy generated by the turbine. Hence net turbine efficiency, which is the actual energy output available will be less than what has been calculated. In most Gas Turbine plants, air compressor is an integral part of Turbine plant.

7.7.3 Reciprocating Engine Systems

This system (Figure 7.12) provides process heat or steam from engine exhaust. The waste heat from engine jacket and lube oil cooler may also be used to provide hot water or hot air. There are, however, limited applications for this.

As these engines can use only fuels like HSD, distillate, residual oils, natural gas, LPG etc. and as they are not economically better than steam/gas turbine, their use is not widespread for co-generation. One more reason for this is the engine maintenance requirement.



7.12 Reciprocating Engine System

7.8 Typical Cogeneration Performance Parameters

The following Table 7.4 gives typical Cogeneration Performance Parameters for different Cogeneration Packages giving heat rate, overall efficiencies etc.

Table 7.4: Typical Cogeneration Performance Parameters

Prime Mover in Cogen. Package	Nominal Range (Electrical)	Electrical Generation Heat Rate (kcal / kWh)	Efficiencies, %		
			Electrical Conversion	Thermal Recovery	Overall Cogeneration
Smaller Reciprocating Engines	10 – 500 kW	2650 - 6300	20-32	50	74-82

Larger Reciprocating Engines	500 – 3000 kW	2400 - 3275	26-36	50	76-86
Diesel Engines	10-3000 kW	2770 - 3775	23-38	50	73-88
Smaller Gas Turbines	800-10000 kW	2770-3525	24-31	50	74-81
Larger Gas Turbines	10-20 MW	2770-3275	26-31	50	78-81
Steam Turbines	10-100 MW	2520-5040	17-34	-	-
Note: Adapted from Cogeneration Handbook California Energy Commission, 1982					

7.9 Relative Merits of Cogeneration Systems

The following Table 7.5 gives the advantages and disadvantages of various co-generation systems:

Table7.5 Advantages and Disadvantages of Various Cogeneration Systems

Variant	Advantages	Disadvantages
Back pressure	- High fuel efficiency rating	- Little flexibility in design and operation
Steam turbine & fuel firing in boiler	- Simple plant - Well-suited to low quality fuels	- More capital investment - Low fuel efficiency rating - High cooling water demand - More impact on environment - High civil const. cost due to complicated foundations
Gas turbine with waste heat recovery boiler	- Good fuel efficiency - Simple plant - Low civil const. Cost - Less delivery period - Less impact on environment - High flexibility in operation	- Moderate part load efficiency - Limited suitability for low quality fuels
Combined gas & steam turbine with waste heat recovery boiler	- Optimum fuel efficiency rating - Low relative capital cost - Less gestation period - Quick start up & stoppage - Less impact on environment - High flexibility in operation	- Average to moderate part-load efficiency - Limited suitability for low quality fuels
Diesel Engine & waste heat recovery Boiler & cooling water heat exchanger	- Low civil const. Cost due to block foundations & least no. of auxiliaries - High Power efficiency - Better suitability as stand by power source	- Low overall efficiency - Limited suitability for low quality fuels - Availability of low temperature steam - Highly maintenance prone.

7.10 Steam Turbine Efficiency

Turbine efficiency

Several terms are in use with respect to turbine efficiency.

- "Turbine cylinder efficiency is given as ratio of actual enthalpy drop and isentropic enthalpy drop, as a percentage."
- Performance of steam turbine is also assessed by a pressure survey to establish extent of performance deviation.
- Performance of steam turbine is also expressed as heat rate, which is the quantity of heat in kcal or KJ required to generate 1 kWh of electrical power output."
- "Turbine efficiency is the ratio useful heat and power output, to the heat input to the turbine in Kcal or KJ, expressed as a percentage."
- Performance of steam turbines is also expressed in terms of Theoretical Steam Rate (TSR) and Actual Steam Rate (ASR), to generate one kWh of electric power. TSR and ASR can be determined from the power generation and the steam input data.

Methods of Turbine performance monitoring

Cylinder efficiency

The ratio of the actual heat drop in a cylinder to the isentropic heat drop is cylinder efficiency. As illustrated in Figure 7.13, the parameters at Turbine inlet and exhaust are measured, to arrive at effective heat drop A.

If there were no friction, turbulence, windage and other losess, the expansion would be isentropic and heat drop would be B. The cylinder efficiency is thus expressed as A/B in percentage.

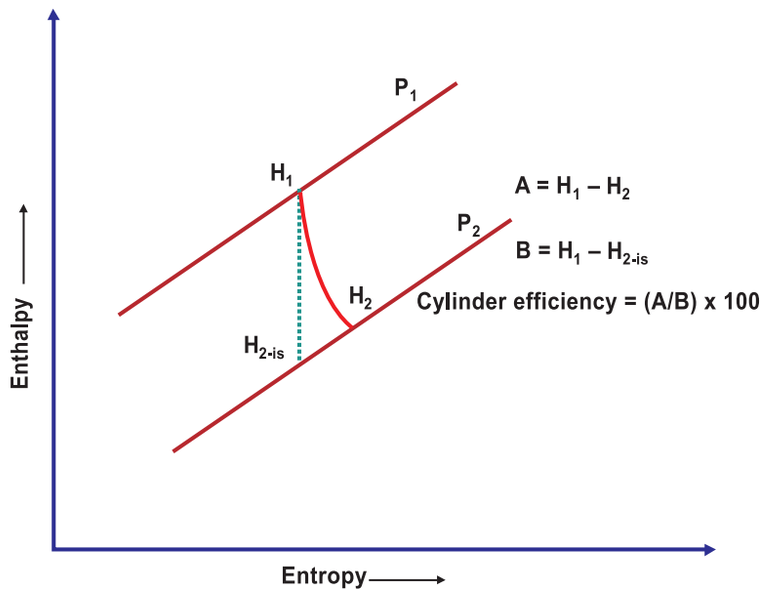


Fig 7.13 Actual & Isentropic Heat Drop in a Turbine

Turbine Pressure survey:

A very useful assessment of internal condition of a turbine is achieved by carrying out a pressure survey. Pressure is noted at various tapping points across turbine expansion path and this pressure profile is validated against the same at design conditions, to infer internal condition of turbine. It is important that pressure gauges used are accurate and that all readings are obtained while load is steady. Pressure survey should be carried out on each running machine frequently.

Turbine Heat rate:

Heat rate is the heat input to turbine, needed to produce 1 kWh of electricity. Turbine heat rate is expressed in kJ/kWh. The inverse relation between heat rate and efficiency is applicable only to a power plant, since all the input energy is deployed for power generation alone.

Notes on cogeneration efficiency

- Cogeneration efficiency, (being useful heat output by heat input) is not necessarily inverse of heat rate, as in case of a typical thermal plant, since the useful heat output to process is not accounted in heat rate calculation.
- Back pressure steam turbines offer better system efficiency up to 85% since all the exhaust heat in steam is used in process, as useful heat.
- Extraction cum back pressure and extraction cum condensing turbines are deployed where flexibility for meeting various steam and power demands is called for. As the steam quantity passed through condenser increases, the heat rate becomes higher, since loss of latent heat in condenser is significant. The extraction condensing cogeneration systems, on this account, need more heat input (i.e. higher heat rate) and therefore are less efficient then back pressure type.
- Fully condensing turbines are used only by utility plants, and they have a huge share of heat input lost through condenser & cooling tower system.
- In Industrial Cogeneration systems, back up pressure or extraction back pressure turbines are preferred for economy, efficiency and flexibility.

7.11 Cogeneration Heat Rate and Efficiency Assessment -Illustrative Case

Co-generation Plant Features

Boiler

Coal Fired FBC Water Tube Type, 35 TPH, 66 ata, 505 ± 5 ^{O}C and 1537 m^2 Heating Surface Area, 83% Boiler efficiency

Boiler feed water temperature = 60 ^{O}C. Feed water enthalpy = 251 kJ/kg

Turbine

Bleed Cum Extraction Cum Condensing, 6000 kW Alternator, 63 kg / cm^2 (g) Steam inlet pressure at 500 ^OC and 34800 kg/hr Flow, Extraction Pressure of 8 kg/cm^2, Extraction Flow of 13470 kg/hr, Bleed extraction Pressure of 1.23 kg/cm^2 and Flow of 3550 kg/hr for deaerator, Exhaust Pressure of 0.10 ata, Turbine rpm 8280 and Alternator rpm 1500.

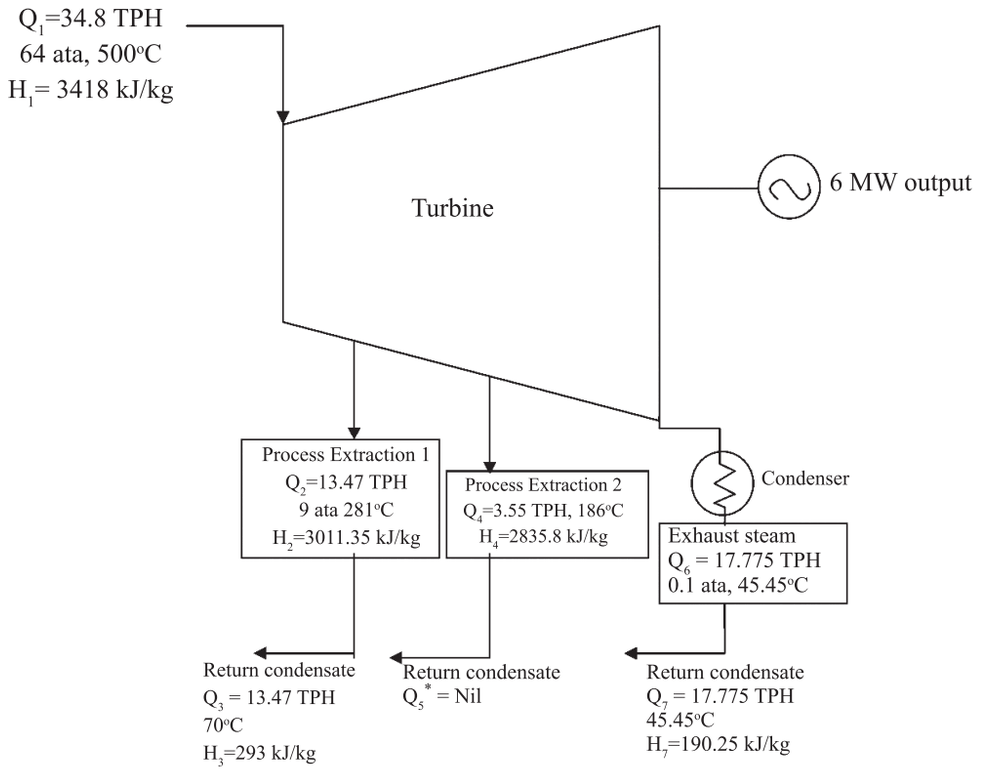
Surface Condenser

Fixed Tube, Two Pass (Divided Type) Condenser, Steam Load 22000 kg/hr @ 0.095 ata and 40^OC, CW Flow 1514 m^3/hr, HT Area 649.2 m^2.

The performance of cogeneration plant at various settings is as follows:

S. No.	Item Reference	Case 1
1.0	Operating conditions	6 MW output & normal flow to process
2.0	**Operating Parameters**	
2.1	Steam condition at TSV	34.8 TPH 64 ATA 500^OC (3418 kJ/kg)
2.2	Steam conditions to process extraction	13.47 TPH 9 ATA 281^OC (3011.35 kJ/kg)
2.3	Steam condition to deaerator (Bleed)	3.55 TPH 2.26 ATA 186 ^OC (2835.8 kJ/kg)
2.4	Exhaust steam Conditions	17.775 TPH 0.1 ATA 45.45^OC (190.25 kJ/kg)

Case 1: Turbine heat rate with 6 MW output, normal flow to process



*Q_5: Direct injection of steam to process

(1kWh= 3600 kJ)

$$\text{Heat to Power Ratio} = \frac{\{[(Q_2 \times (H_2 - H_3)) + (Q_4 \times H_4)]/3600\}\text{kW}}{6000\,\text{kW}}$$

$$= \frac{\{[(13470 \times (3011.35 - 293)) + (3550 \times 2835.8)]/3600\}\text{kW}}{6000\,\text{kW}}$$

$$= \frac{\{(36616174.5 + 10067090)/3600\}\text{kW}}{6000\,\text{kW}}$$

$$= \frac{12967.6\text{kW}}{6000\,\text{kW}}$$

$$= 2.16$$

$$\textit{Energy Utilisation Factor, EUF} = \frac{\textit{Electrical output, } Q_e + \textit{Thermal ouput, } Q_{th}}{\textit{Fuel heat input, } Q_f}$$

Q_e = 6000 x 860 = 5160 Mcal/hr

$$Q_{th} = [(13470 \times (3011.35 - 293)) + (3550 \times 2835.8)]/4.187$$

= 11150 Mcal/hr

$$Q_f = \frac{\{(34800 \times (3418 - 251))/4.187\}}{0.83}$$

= 31714 Mcal/hr

$$\text{Energy Utilisation Factor (EUF)} = \frac{\mathbf{5160 + 11150}}{\mathbf{31714}}$$

= 51.43%

7.12 Trigeneration

Trigeneration refers to simultaneous generation of steam (heat), power and refrigeration through integrated systems. Industries requiring electricity, steam and cooling such as food processing and cold storages find the concepts of tri-generation very attractive.

A combined cycle trigeneration plant could typically consist of a gas turbine generator, waste heat recovery boiler, steam turbine, generator, and absorption chiller, to meet 100% of the facility energy needs. Whenever the power is surplus, it is sold to the grid.

An illustrative trigeneration system schematic is presented in Figure 7.14.

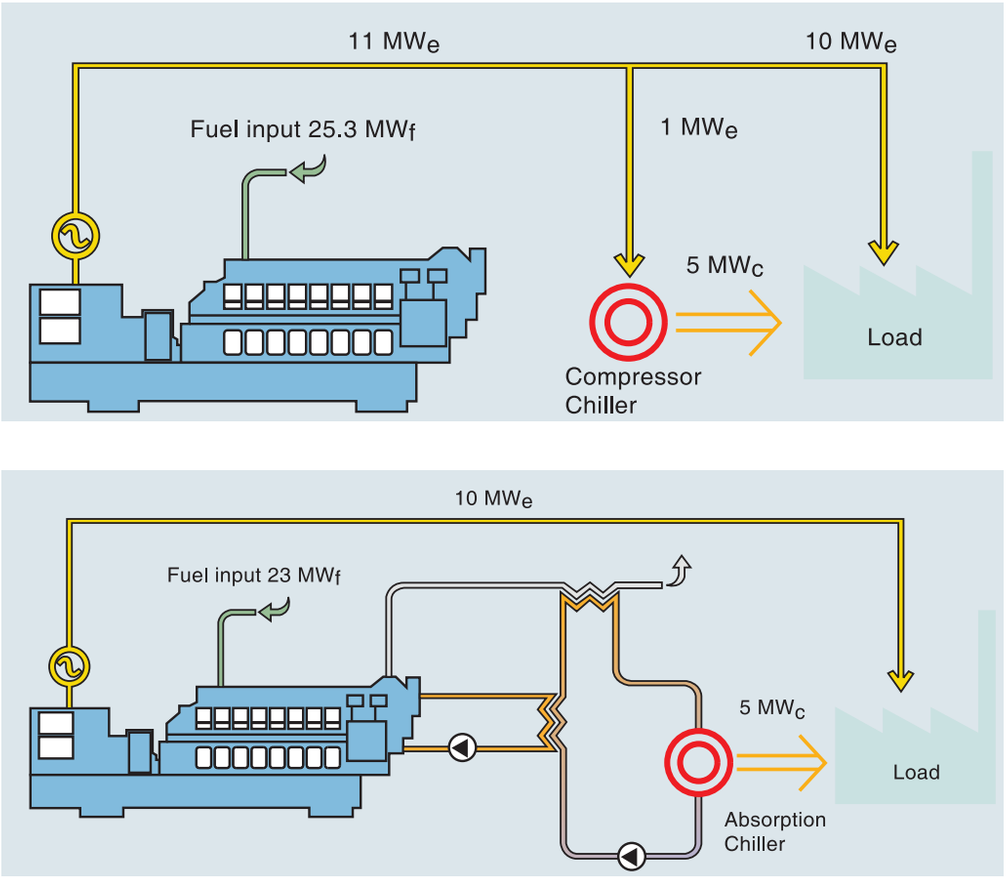


Figure 7.14: Trigeneration: Power, Heating and Cooling

7.13 Microturbine

The Micro steam Turbine Power System is a compact, efficient power system that generates electricity from pressure energy previously wasted in the steam pressure reducing valves.

Figure 7.15a PRV

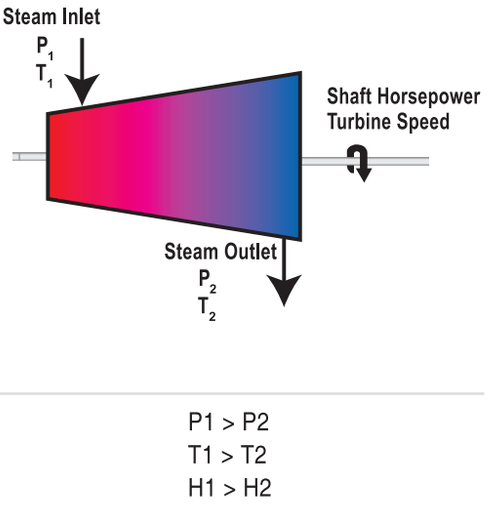


Figure 7.15b Micro turbine

In a typical steam system, a Pressure Reducing Valve (PRV) reduces the steam pressure from P1 to P2 (Figure 7.15a). This pressure reduction happens in such a manner that the total energy content (enthalpy) does not change (H1 = H2) and no shaft work is done.

On the other hand, when steam goes through a steam turbine, it expands and the steam pressure reduces from P1 to P2 (Figure 7.15b). The steam turbine produces shaft horsepower and as a result, the steam exit energy content (enthalpy) is lower when compared to the PRV case.

Steam is used for heating purposes in the plant. The process heat duty is fixed by the plant demand. Since the steam supplied to the process has a lower enthalpy, an additional amount of steam is required to ensure the same available heat duty. This additional amount of steam has an associated cost. Hence, power generated from a backpressure steam turbine is **not** free.

Nevertheless, using a backpressure steam turbine can improve the overall plant and global energy efficiency and more importantly, it can reduce total operating costs.

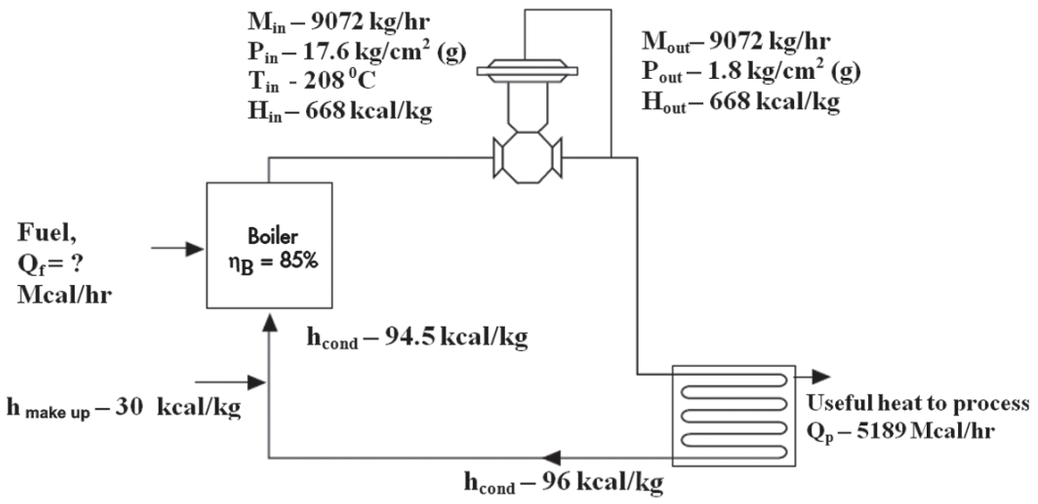
Example : Pressure reducing valve versus back pressure turbine

A chemical plant generates steam at 17.6 kg/cm^2 (g) which is further reduced to 1.8 kg/cm^2 (g) through a pressure reducing valve, for utilization in the process. The process requires 5189 MCal/hr of heat. The plant has planned to replace the PRV with a back pressure steam turbine.

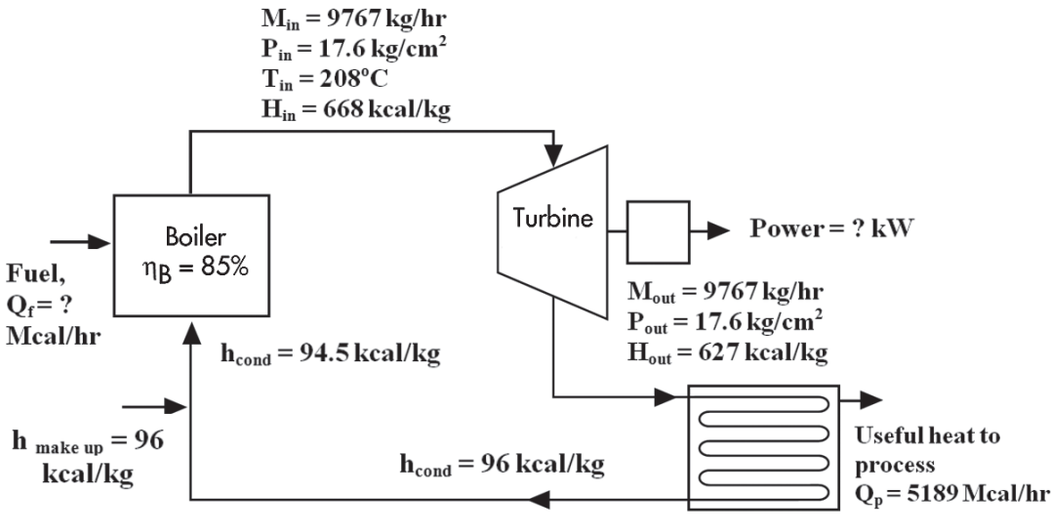
Calculate the net savings per annum if the plant operates 8000 hrs/annum with the following data and the data provided in the schematic diagrams.

Calorific value of fuel	- 10,000 kcal/kg,
Fuel cost	- Rs. 25,000/Tonne,
Grid cost of electricity charge	- Rs. 6/kWh.
Gear box efficiency	- 98%
Alternator efficiency	- 98%
Mechanical efficiency of turbine	- 97%

a) Boiler and PRV operation



b) Boiler and Micro turbine operation



Solution:

Fuel consumption with PRV operation

$$= \frac{9072 \times (668 - 94.5)}{0.85 \times 10000}$$

= 612.1 kg/hr

Fuel consumption with steam turbine

$$= \frac{9767 \times (668 - 94.5)}{0.85 \times 10000}$$

= 659 kg/hr

Additional cost of fuel

$$= (659 - 612.1) \times Rs.25.0 / kg$$

= Rs. 1172.5 /hr

Power generated

$$= \frac{9767 \times (668 - 627) \times 0.98 \times 0.98 \times 0.97}{860}$$

= 434 kW

Grid electricity savings

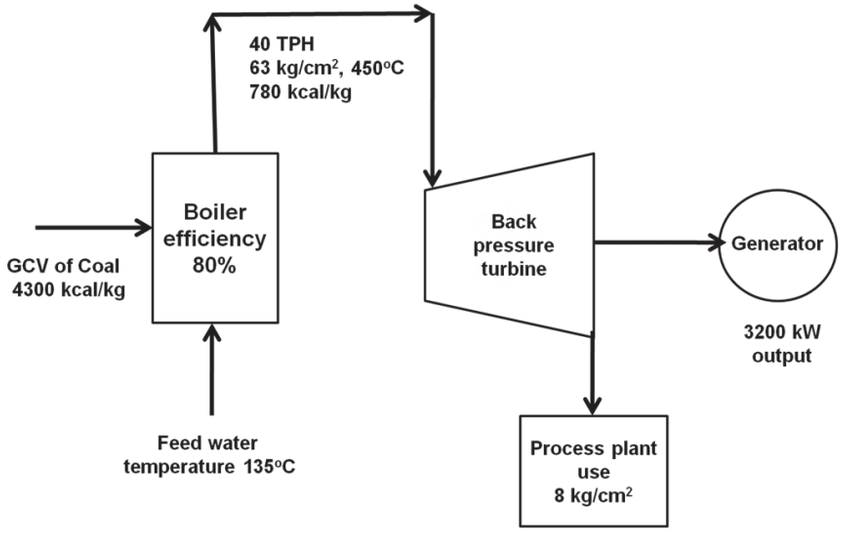
= 434 x 6 = Rs. 2604/hr

Net Savings

= (2604 – 1172.5) x 8000 = Rs. 114.5 Lakh/annum

Solved Example:

The schematic of a back pressure steam turbine cogeneration system of process plant operating round the clock with operating data is depicted below.



If the steam requirement of the process is to be increased to 44TPH which can be met by the existing boiler through the back pressure turbine,

a) Find out the reduction in cost of electrical energy drawn from the grid per day due to additional power generation ,assuming the same steam to power recovery as in the existing case and at a grid electricity cost of Rs.7/kWh, Aux power remains the same.

b) Also find out the additional coal requirement per day?

Ans :

a) Present steam to power recovery ratio = 40,000 kg/hr / 3,200 kWh
= 12.5 kg/kWh

Additional steam generation = 4TPH
Additional power generation potential = 4000 kg/hr /12.5 kg/kWh
= 320 kW

Daily saving due to additional power generation = 320 kW x 24 hr x Rs.7
= Rs 53,760

Additional coal requirement per hour = 4000 x (780-135) / (0.8 x 4300)
= 750 kg/hr

b) Additional coal requirement per day = 750 kg/hr x 24
= 18,000 kg/day = **18 Ton/day**

QUESTIONS

Objective Type Questions

1. A cycle which generates first electricity or mechanical power is called

   a) bottoming cycle b) topping cycle c) combined cycle d) cogeneration cycle

2. A power plant which uses first a gas turbine followed by steam turbine for power generation is called

   a) Rankine cycle b) combined cycle c) Brayton cycle d) bottoming cycle

3. For same inlet conditions of the steam which of the following will generate the maximum mechanical power

   a) condensing turbine b) back pressure turbine
   c) extraction-cum-condensing turbine d) extraction-cum-back pressure turbine

4. The equipment having the highest efficiency in case of cogeneration plant is

   a) electric generator b) cooling tower c) steam or gas turbine d) boiler

5. The efficiency of a stand-alone gas turbine without any heat recovery system will be in the range of

   a) 10 –15 % b) 15 –20 % c) 20 – 25 % d) 35 – 40 %

6. An axial compressor is used in conjunction with which of the following

   a) Back pressure steam turbine b) Gas turbine
   c) condensing turbine d) extraction cum condesing turbine

7. A co-generation system is also known as

   a) reversible system b) re-generation system
   c) combined heat and power system d) Brayton cycle system

8. Power is generated from the waste heat of a sponge iron plant. This type of cogeneration is called

   a) topping cycle b) bottoming cycle
   c) gas turbine cycle d) none of the above.

9. Air compressor alone consumes about _____ of the energy generated in a gas turbine

   a) 20-30% b) 30-40% c) 40-50% d) 50-60%

10. A paper plant needs steam at 3 bar and 10 bar in addition to electric power. The most suitable cogeneration choice among the following will be

    a) condensing turbine b) back pressure turbine
    c) extraction cum back pressure turbine d) bottoming cycle

Short Type Questions

S-1 **Match the following:**

Cogeneration system	**Typical range of heat to power ratio**
a) Back pressure steam turbine	1) 4.0 - 14.3
b) Extraction-condensing steam turbine	2) 1.1 – 2.5
c) Gas turbine	3) 1.0 – 1.7
d) Combined cycle	4) 1.3 – 2.0
e) Reciprocating engine	5) 2.0 – 10.0

S-2	Draw a sketch of an extraction-condensing turbine cogeneration system.
S-3	What is heat-to-power ratio of cogeneration system and state its importance?
S-4	Sketch the schematic of "Back Pressure Turbine" and "Extraction Condensing Turbine" Cogeneration systems
S-5	Draw a schematic diagram of a combined cycle power plant
	Long Type Questions
L-1	Compare the cost of power generation from a gas turbine installed with a 10 TPH waste heat recovery boiler vis-a-vis grid supply The operational data are as under: Capacity of gas turbine : 3000kW Auxiliary power consumption : 1% of electricity generated Operating hours per annum : 8000 Plant load factor : 90% Heat rate : 3050kcal/kWh Calorific value of natural gas : 9500kcal/sm^3 Cost of gas : Rs.6000/1000 sm^3 Cost of capital of gas turbine and boiler and maintenance of W.H.R charges per annum : Rs.400 lakhs Cost of electric power from grid (Demand and energy charges) : Rs.4.5/kWh
L-2	For a backpressure steam turbine with the following operating data, evaluate the heat to power ratio (kW_{TH}/ kW_E) if the turbine and generator efficiencies are 90% and 92% respectively.

Steam inlet conditions to the back pressure turbine	Steam outlet conditions of the back pressure turbine
P = 5.52 MPa T = 538ºC h = 3515.3 kJ/kg Q = 72 MT/hr	P = 1.10 Mpa T = 288ºC h = 3022.0 kJ/kg h_f = 781.7 kJ/kg

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   www.energysolutionscenter.org/distgen/AppGuid (microturbines)

Chapter 8

Waste Heat Recovery

Learning Objectives

In this chapter you will learn about

- Waste heat – quality and quantity
- Waste heat sources and grades
- High, medium and low temperature heat recovery
- Evolving waste heat recovery scheme
- Types of waste heat recovery devices and applications

8. WASTE HEAT RECOVERY

Classification, Advantages and Applications, Commercially Viable Waste Heat Recovery Devices, Saving Potential.

8.1 Introduction

Waste heat is heat, which is generated in a process by way of fuel combustion or chemical reaction, and then "dumped" into the environment even though it could still be reused for some useful and economic purpose. The essential quality of heat is not the amount but rather its "value". The strategy of how to recover this heat depends in part on the temperature of the waste heat gases and the economics involved.

Large quantity of hot flue gases is generated from Boilers, Kilns, Ovens and Furnaces. If some of this waste heat could be recovered, a considerable amount of primary fuel could be saved. The energy lost in waste gases cannot be fully recovered. However, much of the heat could be recovered and loss minimized by adopting following measures as outlined in this chapter.

Heat Losses –Quality

Depending upon the type of process, waste heat can be rejected at virtually any temperature from that of chilled cooling water to high temperature waste gases from an industrial furnace or kiln. Usually higher the temperature, higher the quality and more cost effective is the heat recovery. In any study of waste heat recovery, it is absolutely necessary that there should be some use for the recovered heat. Typical examples of use would be preheating of combustion air, space heating, or pre-heating boiler feed water or process water. With high temperature heat recovery, a cascade system of waste heat recovery may be practiced to ensure that the maximum amount of heat is recovered at the highest potential. An example of this technique of waste heat recovery would be where the high temperature stage was used for air pre-heating and the low temperature stage used for process feed water heating or steam raising.

Heat Losses – Quantity

In any heat recovery situation it is essential to know the amount of heat recoverable and also how it can be used. An example of the availability of waste heat is given below:

- **Heat recovery from heat treatment furnace**

  In a heat treatment furnace, the exhaust gases are leaving the furnace at 900°C at the rate of 2100 m^3/hour. The total heat recoverable at 180°C final exhaust can be calculated as

  $Q = V \times \rho \times C_p \times \Delta T$

Where,

Q is the heat content in kcal

V is the flow rate of the substance in m^3/hr

ρ is density of the flue gas in kg/m^3

C_p is the specific heat of the substance in kcal/kg ^oC

ΔT is the temperature difference in ^oC

Cp (Specific heat of flue gas) = 0.24 kcal/kg/^oC

Heat available (Q) = 2100 x 1.19 x 0.24 x (900-180) = 4,31,827 kcal/hr

By installing a recuperator, this heat can be recovered to pre-heat the combustion air. The fuel savings would be 33% (@ 1% fuel reduction for every 22^{o}C reduction in temperature of flue gas).

8.2 Classification and Application

In considering the potential for heat recovery, it is useful to note all the possibilities, and grade the waste heat in terms of potential value as shown in the following Table 8.1

Table 8.1 Waste Source and Quality

S. No.	Source	Quality
1.	Heat in flue gases.	The higher the temperature, the greater the potential value for heat recovery
2.	Heat in vapour streams.	As above but when condensed, latent heat also recoverable.
3	Convective and radiant heat lost from exterior of equipment	Low grade – if collected may be used for space heating or air preheats.
4.	Heat losses in cooling water.	Low grade – useful gains if heat is exchanged with incoming fresh water.
5.	Heat losses in providing chilled water or in the disposal of chilled water.	a) High grade if it can be utilized to reduce demand for refrigeration. b) Low grade if refrigeration unit used as a form of heat pump.
6.	Heat stored in products leaving the process	Quality depends upon temperature.
7.	Heat in gaseous and liquid effluents leaving process.	Poor if heavily contaminated and thus requiring alloy heat exchanger.

High Temperature Heat Recovery

The following Table 8.2 gives temperatures of waste gases from industrial process equipment in the high temperature range. All of these results from direct fuel fired processes.

Table 8.2 Typical Waste Heat Temperature at High Temperature Range from Various Sources	
Types of Device	**Temperature, °C**
Nickel refining furnace	1370 –1650
Aluminium refining furnace	650-760
Zinc refining furnace	760-1100
Copper refining furnace	760- 815
Steel heating furnaces	925-1050
Copper reverberatory furnace	900-1100
Open hearth furnace	650-700
Cement kiln (Dry process)	620- 730
Glass melting furnace	1000-1550
Hydrogen plants	650-1000
Solid waste incinerators	650-1000
Fume incinerators	650-1450

Medium Temperature Heat Recovery

The following Table 8.3 gives the temperatures of waste gases from process equipment in the medium temperature range. Most of the waste heat in this temperature range comes from the exhaust of directly fired process units.

Table 8.3 Typical Waste Heat Temperature at Medium Temperature Range From Various Sources	
Type of Device	**Temperature, °C**
Steam boiler exhausts	230-480
Gas turbine exhausts	370-540
Reciprocating engine exhausts	315-600
Reciprocating engine exhausts (turbo charged)	230- 370
Heat treating furnaces	425 - 650
Drying and baking ovens	230 - 600
Catalytic crackers	425 - 650
Annealing furnace cooling systems	425 - 650

Low Temperature Heat Recovery

The following Table 8.4 lists some heat sources in the low temperature range. In this range it is usually not practical to extract work from the source, though steam production may not be completely excluded if there is a need for low-pressure steam. Low temperature waste heat may be useful in a supplementary way for preheating purposes.

Table 8.4 Typical Waste Heat Temperature at Low Temperature Range from Various Sources	
Source	**Temperature, °C**
Process steam condensate	55-88
Cooling water from:	
Furnace doors	32-55
Bearings	32-88
Welding machines	32-88
Injection molding machines	32-88
Annealing furnaces	66-230
Forming dies	27-88
Air compressors	27-50
Pumps	27-88
Internal combustion engines	66-120
Air conditioning and refrigeration condensers	32–43
Liquid still condensers	32-88
Drying, baking and curing ovens	93-230
Hot processed liquids	32-232
Hot processed solids	93-232

8.3 Benefits of Waste Heat Recovery

Benefits of 'waste heat recovery' can be broadly classified in two categories:

Direct Benefits:

Recovery of waste heat has a direct effect on the efficiency of the process. This is reflected by reduction in the utility consumption & costs, and process cost.

Indirect Benefits:

a) **Reduction in pollution**: A number of toxic combustible wastes such as carbon monoxide gas, sour gas, carbon black off gases, oil sludge, Acrylonitrile and other plastic chemicals etc, releasing to atmosphere if/when burnt in the incinerators serves dual purpose i.e. recovers heat and reduces the environmental pollution levels.

b) **Reduction in equipment sizes**: Waste heat recovery reduces the fuel consumption, which leads to reduction in the flue gas produced. This results in reduction in equipment sizes of all flue gas handling equipments such as fans, stacks, ducts, burners, etc.

c) **Reduction in auxiliary energy consumption**: Reduction in equipment sizes gives additional benefits in the form of reduction in auxiliary energy consumption like electricity for fans, pumps etc..

8.4 Development of a Waste Heat Recovery System

Understanding the process

Understanding the process is essential for development of Waste Heat Recovery system. This can be accomplished by reviewing the process flow sheets, layout diagrams, piping isometrics, electrical and instrumentation cable ducting etc. Detail review of these documents will help in identifying:

a) Sources and uses of waste heat
b) Upset conditions occurring in the plant due to heat recovery
c) Availability of space
d) Any other constraint, such as dew point occurring in an equipments etc.

After identifying source of waste heat and the possible use of it, the next step is to select suitable heat recovery system and equipments to recover and utilise the same.

Economic Evaluation of Waste Heat Recovery System

It is necessary to evaluate the selected waste heat recovery system on the basis of financial analysis such as investment, depreciation, payback period, rate of return etc. In addition the advice of experienced consultants and suppliers must be obtained for rational decision.

Next section gives a brief description of common heat recovery devices available commercially and its typical industrial applications.

8.5 Commercial Waste Heat Recovery Devices

Recuperators

In a recuperator, heat exchange takes place between the flue gases and the air through metallic or ceramic walls. Duct or tubes carry the air for combustion to be pre-heated, the other side contains the waste heat stream. A recuperator for recovering waste heat from flue gases is shown in Figure 8.1.

The simplest configuration for a recuperator is the metallic radiation recuperator, which consists of two concentric lengths of metal tubing as shown in Figure 8.2.

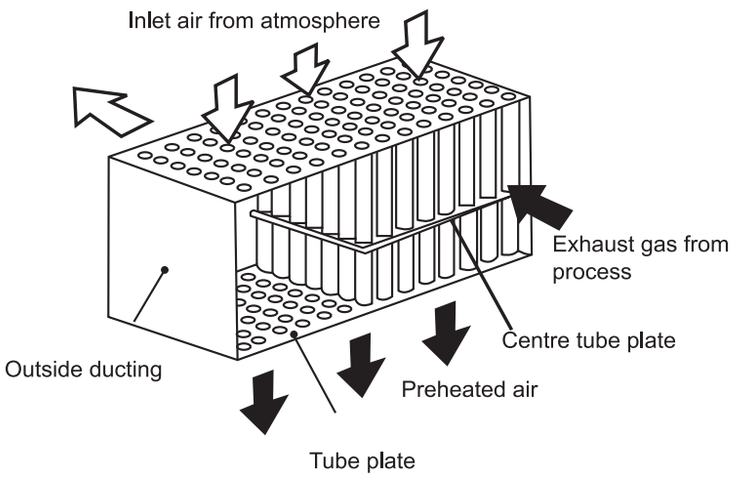

Figure 8.1 WHR using Recuperator

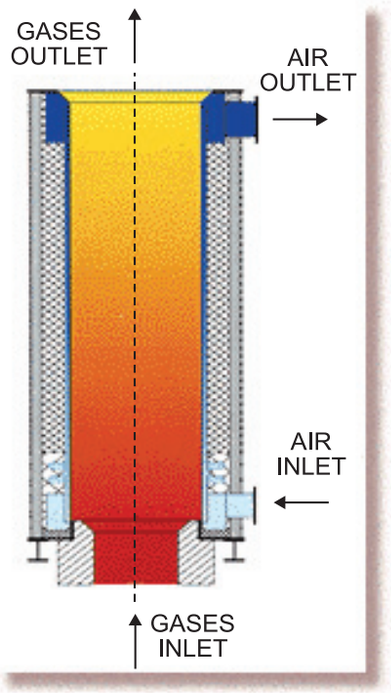

Figure 8.2 Metallic Radiation Recuperator

The inner tube carries the hot exhaust gases while the external annulus carries the combustion air from the atmosphere to the air inlets of the furnace burners. The hot gases are cooled by the incoming combustion air which now carries additional energy into the combustion chamber. This is energy which does not have to be supplied by the fuel; consequently, less fuel is burned for a given furnace loading. The saving in fuel also means a decrease in combustion air and therefore stack losses are decreased not only by lowering the stack gas temperatures but also by discharging smaller quantities of exhaust gas. The radiation recuperator gets its name from the fact that a substantial portion of the heat transfer from the hot gases to the surface of the inner tube takes place by radiative heat transfer. The cold air in the annuals, however, is almost transparent to infrared radiation so that only convection heat transfer takes place to the incoming air. As shown in the diagram, the two gas flows are usually parallel, although the configuration would be simpler and the heat transfer more efficient if the flows were opposed in direction (or counterflow). The reason for the use of parallel flow is that recuperators frequently serve the additional function of cooling the duct carrying away the exhaust gases and consequently extending its service life.

A second common configuration for recuperators is called the tube type or convective recuperator. As seen in the Figure 8.3, the hot gases are carried through a number of parallel small diameter tubes, while the incoming air to be heated enters a shell surrounding the tubes and passes over the hot tubes one or more times in a direction normal to their axes.

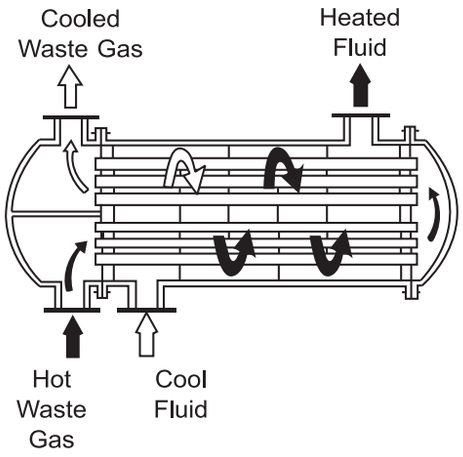

Figure 8.3 Convective Recuperator

If the tubes are baffled to allow the gas to pass over them twice, the heat exchanger is termed a two-pass recuperator; if two baffles are used, a three-pass recuperator, etc. Although baffling increases both the cost of the exchanger and the pressure drop in the combustion air path, it increases the effectiveness of heat exchange. Shell and tube type recuperators are generally more compact and have a higher effectiveness than radiation recuperators, because of the larger heat transfer area made possible through the use of multiple tubes and multiple passes of the gases.

Radiation/Convective Hybrid Recuperator:

For maximum effectiveness of heat transfer, combinations of radiation and convective designs are used, with the high-temperature radiation recuperator being first followed by convection type.

These are more expensive than simple metallic radiation recuperators, but are less bulky. A convective/ radiative hybrid recuperator is shown in Figure 8.4

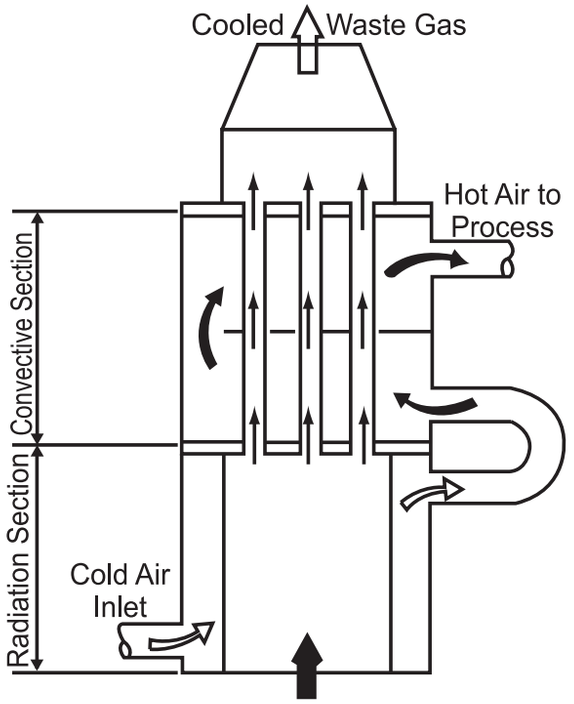

Figure 8.4 Convective Radiative Recuperator

Ceramic Recuperator

The principal limitation on the heat recovery of metal recuperators is the reduced life of the liner at inlet temperatures exceeding 1100 ^{o}C. In order to overcome the temperature limitations of metal recuperators, ceramic tube recuperators have been developed whose materials allow operation on the gas side to 1550 ^{o}C and on the preheated air side to 815 ^{o}C on a more or less practical basis. Early ceramic recuperators were built of tile and joined with furnace cement, and thermal cycling caused cracking of joints and rapid deterioration of the tubes. Later developments introduced various kinds of short silicon carbide tubes which can be joined by flexible seals located in the air headers.

Earlier designs had experienced leakage rates from 8 to 60 percent. The new designs are reported to last two years with air preheat temperatures as high as 700 ^{o}C, with much lower leakage rates.

Regenerator

The Regeneration which is preferable for large capacities has been very widely used in glass and steel melting furnaces. Important relations exist between the size of the regenerator, time between reversals, thickness of brick, conductivity of brick and heat storage ratio of the brick.

In a regenerator, the time between the reversals is an important aspect. Long periods would mean higher thermal storage and hence higher cost. Also long periods of reversal result in lower average temperature of preheat and consequently reduce fuel economy. (Refer Figure 8.5).

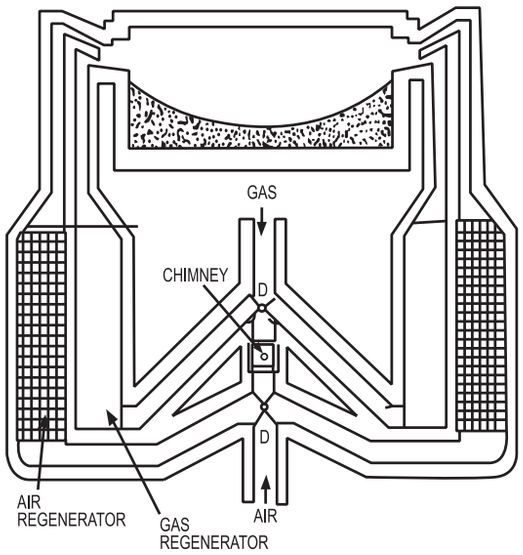

Figure 8.5 Regenerator

Accumulation of dust and slagging on the surfaces reduce efficiency of the heat transfer as the furnace becomes old. Heat losses from the walls of the regenerator and air in leaks during the gas period and out-leaks during air period also reduces the heat transfer.

Heat Wheels

A heat wheel is finding increasing applications in low to medium temperature waste heat recovery systems. Figure 8.6 is a sketch illustrating the application of a heat wheel.

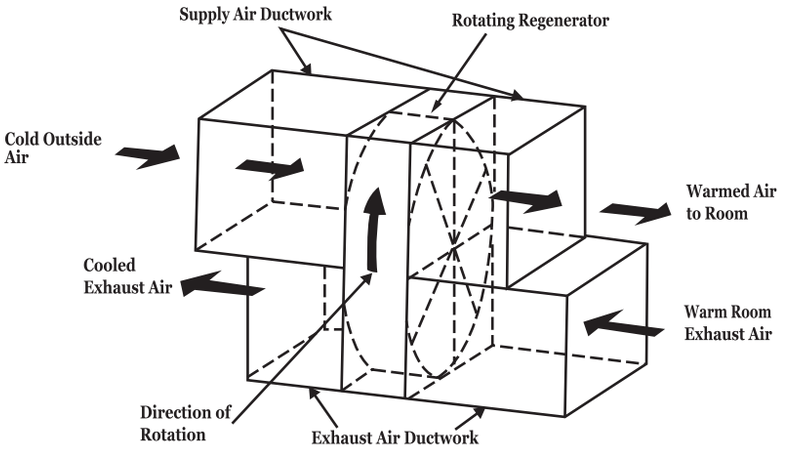

Figure 8.6 Heat Wheel

It is a sizable porous disk, fabricated with material having a fairly high heat capacity, which rotates between two side-by-side ducts: one a cold gas duct, the other a hot gas duct. The axis of the disk is located parallel to, and on the partition between, the two ducts. As the disk slowly rotates, sensible heat (moisture that contains latent heat) is transferred to the disk by the hot air and, as the disk rotates, from the disk to the cold air. The overall efficiency of sensible heat transfer for this kind of regenerator

can be as high as 85 percent. Heat wheels have been built as large as 21 metres in diameter with air capacities up to 1130 m^3 / min.

A variation of the Heat Wheel is the rotary regenerator where the matrix is in a cylinder rotating across the waste gas and air streams. The heat or energy recovery wheel is a rotary gas heat regenerator, which can transfer heat from exhaust to incoming gases.

Its main area of application is where heat exchange between large masses of air having small temperature differences is required. Heating and ventilation systems and recovery of heat from dryer exhaust air are typical applications.

Case Example

A rotary heat regenerator was installed on a two colour printing press to recover some of the heat, which had been previously dissipated to the atmosphere, and used for drying stage of the process. The outlet exhaust temperature before heat recovery was often in excess of 100°C. After heat recovery the temperature was 35°C. Percentage heat recovery was 55% and payback on the investment was estimated to be about 18 months. Cross contamination of the fresh air from the solvent in the exhaust gases was at a very acceptable level.

Case Example

A ceramic firm installed a heat wheel on the preheating zone of a tunnel kiln where 7500 m^3/hour of hot gas at 300°C was being rejected to the atmosphere. The result was that the flue gas temperature was reduced to 150°C and the fresh air drawn from the top of the kiln was preheated to 155°C. The burner previously used for providing the preheated air was no longer required. The capital cost of the equipment was recovered in less than 12 months.

Heat Pipe

A heat pipe can transfer up to 100 times more thermal energy than copper, the best known conductor. In other words, heat pipe is a thermal energy absorbing and transferring system and have no moving parts and hence require minimum maintenance.

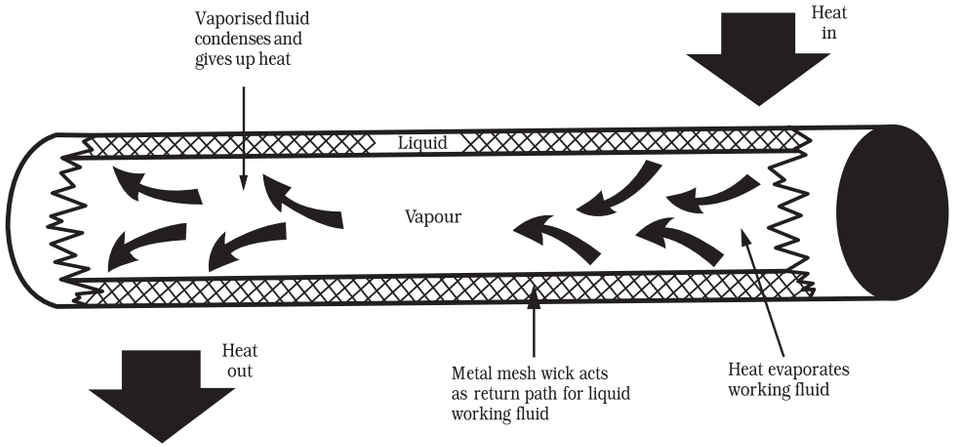

Figure 8.7 Heat Pipe

The Heat Pipe comprises of three elements – a sealed container, a capillary wick structure and a working fluid. The capillary wick structure is integrally fabricated into the interior surface of the container tube and sealed under vacuum. Thermal energy applied to the external surface of the heat pipe is in equilibrium with its own vapour as the container tube is sealed under vacuum. Thermal energy applied to the external surface of the heat pipe causes the working fluid near the surface to evaporate instantaneously. Vapour thus formed absorbs the latent heat of vapourisation and this part of the heat pipe becomes an evaporator region. The vapour then travels to the other end the pipe where the thermal energy is removed causing the vapour to condense into liquid again, thereby giving up the latent heat of the condensation. This part of the heat pipe works as the condenser region. The condensed liquid then flows back to the evaporated region. A figure of Heat pipe is shown in Figure 8.7

Performance and Advantage

The heat pipe exchanger (HPHE) is a lightweight compact heat recovery system. It virtually does not need mechanical maintenance, as there are no moving parts to wear out. It does not need input power for its operation and is free from cooling water and lubrication systems. It also lowers the fan horsepower requirement and increases the overall thermal efficiency of the system. The heat pipe heat recovery systems are capable of operating at 315^{o}C with 60% to 80% heat recovery capability.

Typical Application

The heat pipes are used in following industrial applications:

a. Process to Space Heating: The heat pipe heat exchanger transfers the thermal energy from process exhaust for building heating. The preheated air can be blended if required. The requirement of additional heating equipment to deliver heated make up air is drastically reduced or eliminated.

b. Process to Process: The heat pipe heat exchangers recover waste thermal energy from the process exhaust and transfer this energy to the incoming process air. The incoming air thus become warm and can be used for the same process/other processes and reduces process energy consumption.

c. HVAC Applications:

   Cooling: Heat pipe heat exchangers precools the building make up air in summer and thus reduces the total tons of refrigeration, apart from the operational saving of the cooling system. Thermal energy is supply recovered from the cool exhaust and transferred to the hot supply make up air.
   Heating: The above process is reversed during winter to preheat the make up air.

The other applications in industries are:

- Preheating of boiler combustion air
- Recovery of Waste heat from furnaces

- Reheating of fresh air for hot air driers
- Recovery of waste heat from catalytic deodorizing equipment
- Reuse of Furnace waste heat as heat source for other oven
- Cooling of closed rooms with outside air
- Preheating of boiler feed water with waste heat recovery from flue gases in the heat pipe economizers.
- Drying, curing and baking ovens
- Waste steam reclamation
- Brick kilns (secondary recovery)
- Reverberatory furnaces (secondary recovery)
- Heating, ventilating and air-conditioning systems

Case Example

Savings in Hospital Cooling Systems

Volume	140 m^3/min Exhaust
Recovered heat	28225 kcal/hr
Plant capacity reduction	9.33 Tons of Refrigeration
Electricity cost (operation)	Rs. 268/Million kcal (based on 0.8 kW/TR)
Plant capacity reduction cost (Capital)	Rs.12,000/TR
Capital cost savings	Rs. 1,12,000/-
Payback period	16570 hours

Economiser

In case of boiler system, economizer can be provided to utilize the flue gas heat for pre-heating the boiler feed water. On the other hand, in an air pre-heater, the waste heat is used to heat combustion air. In both the cases, there is a corresponding reduction in the fuel requirements of the boiler. An economizer is shown in Figure 8.8.

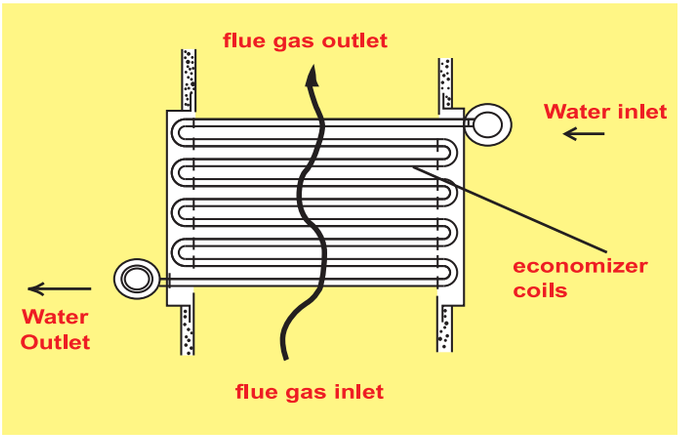

Figure 8.8 Economiser

For every 22^0 C reduction in flue gas temperature by passing through an economiser or a pre-heater, there is 1% saving of fuel in the boiler. In other words, for every 6^0 C rise in feed water temperature through an economiser, or 20^0C rise in combustion air temperature through an air pre-heater, there is 1% saving of fuel in the boiler.

Shell and Tube Heat Exchanger:

When the medium containing waste heat is a liquid or a vapor which heats another liquid, then the shell and tube heat exchanger must be used since both paths must be sealed to contain the pressures of their respective fluids. The shell contains the tube bundle, and usually internal baffles, to direct the fluid in the shell over the tubes in multiple passes. The shell is inherently weaker than the tubes so that the higher-pressure fluid is circulated in the tubes while the lower pressure fluid flows through the shell. When a vapor contains the waste heat, it usually condenses, giving up its latent heat to the liquid being heated. In this application, the vapor is almost invariably contained within the shell. If the reverse is attempted, the condensation of vapors within small diameter parallel tubes causes flow instabilities. Tube and shell heat exchangers are available in a wide range of standard sizes with many combinations of materials for the tubes and shells. A shell and tube heat exchanger is illustrated in Figure 8.9.

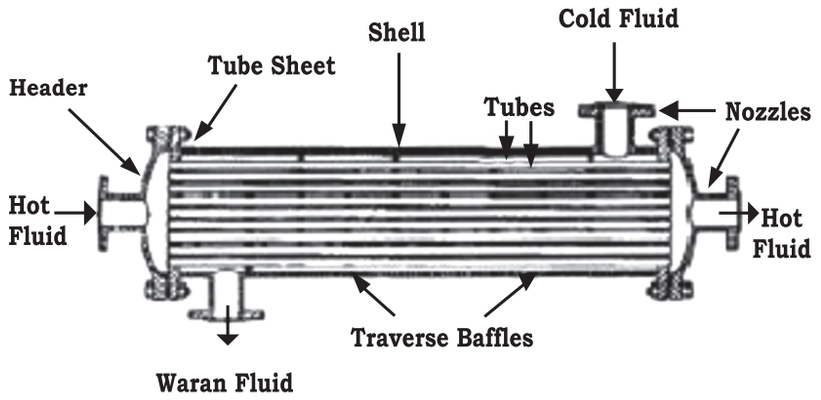

Figure 8.9 Shell & Tube Heat Exchanger

Typical applications of shell and tube heat exchangers include heating liquids with the heat contained by condensates from refrigeration and air-conditioning systems; condensate from process steam; coolants from furnace doors, grates, and pipe supports; coolants from engines, air compressors, bearings, and lubricants; and the condensates from distillation processes.

Plate heat exchanger

The cost of heat exchange surfaces is a major cost factor when the temperature differences are not large. One way of meeting this problem is the plate type heat exchanger, which consists of a series of separate parallel plates forming thin flow pass. Each plate is separated from the next by gaskets and the hot stream passes in parallel through alternative plates whilst the liquid to be heated passes in parallel between the hot plates. To improve heat transfer the plates are corrugated.

Hot liquid passing through a bottom port in the head is permitted to pass upwards between every second plate while cold liquid at the top of the head is permitted to pass downwards between the odd plates. When the directions of hot & cold fluids are opposite, the arrangement is described as counter current. A plate heat exchanger is shown in Figure 8.10.

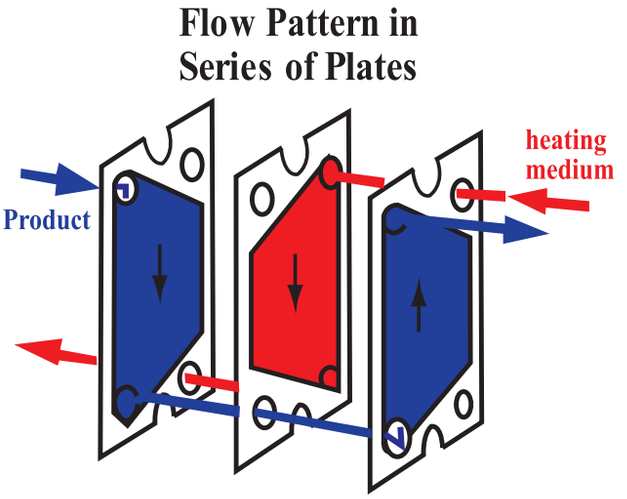

Figure 8.10 Plate Heat Exchanger

Typical industrial applications are:

- Pasteurisation section in milk packaging plant.
- Evaporation plants in food industry.

Run Around Coil Exchanger

It is quite similar in principle to the heat pipe exchanger. The heat from hot fluid is transferred to the colder fluid via an intermediate fluid known as the Heat Transfer Fluid. One coil of this closed loop is installed in the hot stream while the other is in the cold stream. Circulation of this fluid is maintained by means of la circulating pump.

It is more useful when the hot land cold fluids are located far away from each other and are not easily accessible.

Typical industrial applications are heat recovery from ventilation, air conditioning and low temperature heat recovery.

Waste Heat Boilers

Waste heat boilers are ordinarily water tube boilers in which the hot exhaust gases from gas turbines, incinerators, etc., pass over a number of parallel tubes containing water. The water is vaporized in the tubes and collected in a steam drum from which it is drawn off for use as heating or processing steam.

Because the exhaust gases are usually in the medium temperature range and in order to conserve space, a more compact boiler can be produced if the water tubes are finned in order to increase the effective heat transfer area on the gas side. The Figure 8.11 shows a mud drum, a set of tubes over which the hot gases make a double pass, and a steam drum which collects the steam generated above the water surface. The pressure and rate at which the steam is generated depends on the temperature of waste heat. The pressure of a pure vapor in the presence of its liquid is a function of the temperature of the liquid from which it is evaporated. The steam tables tabulate this relationship between saturation pressure and temperature. If the waste heat in the exhaust gases is insufficient for generating the required amount of process steam, auxiliary burners which burn fuel in the waste heat boiler or an

after-burner in the exhaust gases flue are added. Waste heat boilers are built in capacities from 25 m^3 almost 30,000 m^3 / min. of exhaust gas.

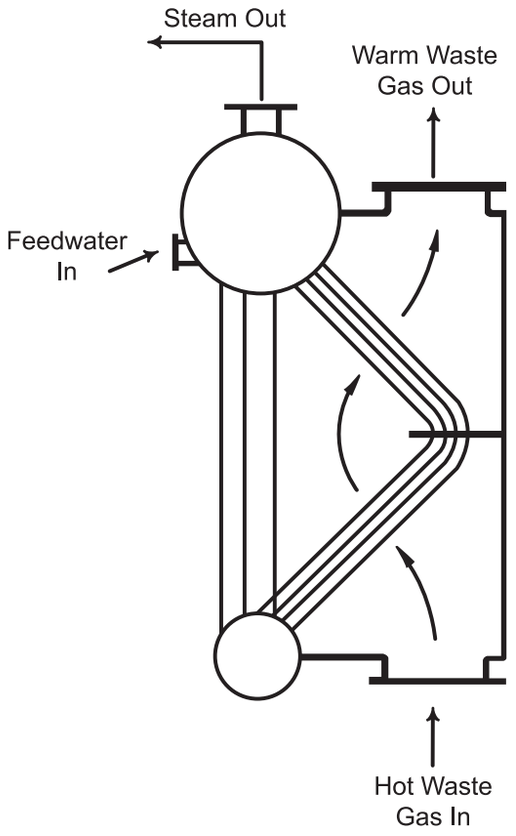

Figure 8.11 Two-Pass Water Tube Waste Heat Recovery Boiler

Typical applications of waste heat boilers are to recover energy from the exhausts of gas turbines, reciprocating engines, incinerators, and furnaces.

Case Example

Gases leaving a carbon black plant rich in carbon monoxide which are vented to the atmosphere.

Equipment Suggested	Carbon monoxide incinerator along with waste heat boiler and steam turbine
Estimated equipment cost	Rs.350 Lakhs
New boiler efficiency	80%
Savings by way of power generated	~Rs.160 Lakhs /annum
Indirect benefits	Reduction in pollution levels

Heat Pumps:

In the various commercial options previously discussed, we find waste heat being transferred from a hot fluid to a fluid at a lower temperature. Heat must flow spontaneously "downhill", that is from a system at high temperature to one at a lower temperature. When energy is repeatedly transferred or

transformed, it becomes less and less available for use. Eventually that energy has such low intensity (resides in a medium at such low temperature) that it is no longer available at all to perform a useful function. It has been taken as a general rule of thumb in industrial operations that fluids with temperatures less than 120°C (or, better, 150°C to provide a safe margin), as limit for waste heat recovery because of the risk of condensation of corrosive liquids. However, as fuel costs continue to rise, even such waste heat can be used economically for space heating and other low temperature applications. It is possible to reverse the direction of spontaneous energy flow by the use of a thermodynamic system known as a heat pump.

The majority of heat pumps work on the principle of the vapour compression cycle. In this cycle, the circulating substance is physically separated from the source (waste heat, with a temperature of T_{in}) and user (heat to be used in the process, T_{out}) streams, and is re-used in a cyclical fashion, therefore called 'closed cycle'. In the heat pump, the following processes take place:

1. In the evaporator the heat is extracted from the heat source to boil the circulating substance;
2. The circulating substance is compressed by the compressor, raising its pressure and temperature; the low temperature vapor is compressed by a compressor, which requires external work. The work done on the vapor raises its pressure and temperature to a level where its energy becomes available for use
3. The heat is delivered to the condenser;
4. The pressure of the circulating substance (working fluid) is reduced back to the evaporator condition in the throttling valve, where the cycle repeats.

The heat pump was developed as a space heating system where low temperature energy from the ambient air, water, or earth is raised to heating system temperatures by doing compression work with an electric motor-driven compressor. The arrangement of a heat pump is shown in Figure 8.12.

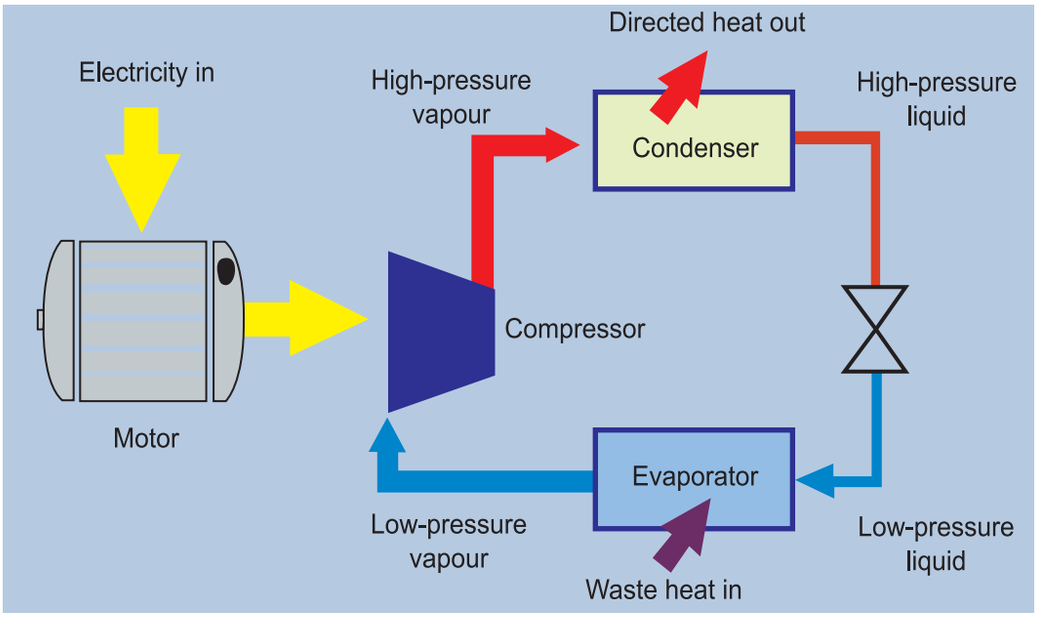

Figure 8.12 Heat pump

The heat pumps have the ability to upgrade heat to a value more than twice that of the energy consumed by the device. The potential for application of heat pump is growing and number of industries have been benefited by recovering low grade waste heat by upgrading it and using it in the main process stream.

Heat pump applications are most promising when both the heating and cooling capabilities can be used in combination. One such example of this is a plastics factory where chilled water from a heat is used to cool injection-moulding machines whilst the heat output from the heat pump is used to provide factory or office heating. Other examples of heat pump installation include product drying, maintaining dry atmosphere for storage and drying compressed air.

Direct Contact Heat Exchanger

Low pressure steam may also be used to preheat the feed water or some other fluid where miscibility is acceptable. This principle is used in Direct Contact Heat Exchanger and finds wide use in a steam generating station. They essentially consist of a number of trays mounted one over the other or packed beds. Steam is supplied below the packing while the cold water is sprayed at the top. The steam is completely condensed in the incoming water thereby heating it. A figure of direct contact heat exchanger is shown in Figure 8.13. Typical application is in the deaerator of a steam generation station.

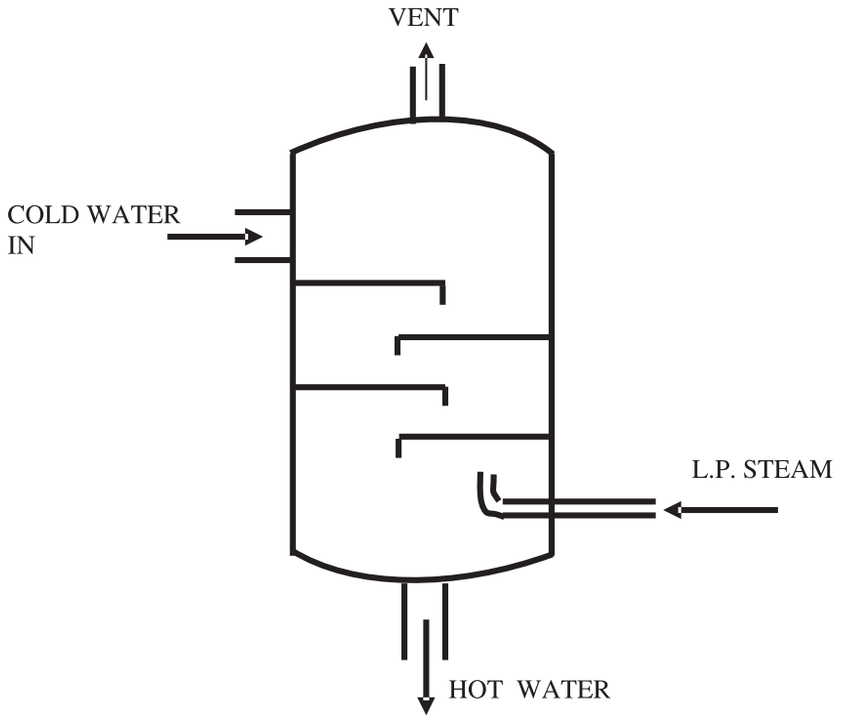

Figure 8.13 Direct Contact Condenser

Solved Example:

In a crude distillation unit of a refinery, furnace is operated to heat 500 m^3/hr of crude oil from 255°C to 360°C by firing 3.4 tons/hr of fuel oil having GCV of 9850 kcal/kg. As an energy conservation measure, the management has installed an air preheater (APH) to reduce the flue gas heat loss. The APH is designed to pre-heat 57 tonnes/hr of combustion air to 195°C.

Calculate the efficiency of the furnace before & after the installation of APH. Consider the following data:

- Specific heat of crude oil = 0.6 kcal/kg°C
- Specific heat of air = 0.24 kcal/kg°C
- Specific gravity of Crude oil = 0.85
- Ambient temperature = 28°C.

Ans:

Before the installation of APH

Heat gain by the crude = 500 x 1000 x 0.85 x 0.6 x (360-255)
= 26775000 kcal/hr

Heat input to the furnace = 3.4 x1000x 9850
= 33490000 kcal/hr

Efficiency of the furnace = 26775000 / 33490000
= 80 %

After the installation of APH

Heat gain by the crude = 500 x 1000 x 0.85 x 0.6 x (360-255)
= 26775000 kcal/hr

Heat gain by Air-preheater = 57 x 1000 x 0.24 x (195-28)
= 2284560 kcal/hr

Heat reduction in the furnace input = Heat gain by Air-preheater
New Heat input to the furnace = 33490000 – 2284560
= 31,205,440 kcal/hr

Efficiency of furnace after installation of APH = 26775000 / 31,205,440
= 85.8 %

QUESTIONS	
Objective Type Questions	
1.	Among which of the following flue gas temperature options, heat recovery equipment will be the least expensive (assuming same material of construction and heat recovered) a) 300°C b) 350°C c) 400°C d) 500°C
2.	Recovery of heat from dryer exhaust air is a typical application of a) waste heat recovery boiler b) heat pump c) heat wheel d) economizer
3.	Which among the following uses a working fluid for heat recovery a) heat pipe b) recuperator c) heat wheel d) regenerator
4.	Economizer in boiler is used to extract heat from flue gases for pre-heating a) blow down b) fuel oil c) air for combustion of fuel d) feed water
5.	The device that upgrades a low temperature heat source to a higher temperature sink is called a) heat pipe b) heat pump c) plate heat exchanger d) economizer
6.	The heat loss rate from a surface is expressed in a) Watt b) Watt/m^2 – ^{0}K c) Watt/m^2 – ^{0}C d) Joules
7.	Regenerators utilizing waste heat are widely used in… a) cement industry b) pulp and paper c) glass melting furnaces d) aluminium
8.	Heat wheels are mostly used in a situation of…. a) high temperature exhaust gases b) heat exchange between large masses of air having small temperature differences c) heat transfer between a liquid and gas d) corrosive gases
9.	A heat pipe can transfer up to ____ times more thermal energy than copper a) 10 b) 20 c) 50 d) 100
10.	Which one is the preferred waste heat recovery system in a large gas turbine? a) economizer b) air pre heater c) boiler d) heat wheel
Short Type Questions	
S-1	What is a ‘heat pipe’? How does it work?
S-2	A firm wants to recover the waste heat in a flue gas stream of 2000 kg/hour. from a furnace. Specific heat of flue gas is 0.25 kcal/kg ^{0}C. 1. Calculate the heat recovered if the heat exchanger has an efficiency of 98% and temperature of flue gas drops from 800^0 C to 250^0 C across the heat exchanger. 2. How many liters of water per hour can be heated by 50 ^{0}C from this waste stream?
S-3	Explain briefly with a neat sketch the operation of a Regenerator for waste heat recovery.

S-4	Describe briefly about 'heat wheels'?
S-5	In a natural gas fired boiler with flue gas temperature of 220 ^{o}C an air preheater is to be installed to recover the waste heat. Name five important parameters to be collected for evaluating the fuel consumption reduction due to waste heat recovery.
Long Type Questions	
L-1	On the topic of waste heat recovery boilers, explain the following: 1. Which are typical applications of waste heat boilers? 2. How do they differ from ordinary steam boilers? 3. In what temperature range do they operate? 4. Is it more energy efficient to generate hot water of 80^{0} C or saturated steam at 6 bar in a waste heat boiler? Explain
L-2	1. Draw the schematics of a heat pump system and explain its operation 2. State two examples of heat pump applications

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Chapter 9

Heat Exchanger

Learning Objectives

In this chapter you will learn about

- Heat exchanger concept
- Heat exchanger types
- Heat exchanger performance
- Process integration and pinch analysis
- Heat exchange equipment

9. HEAT EXCHANGERS

Heat Transfer Basics, Heat Exchangers, Types, Condensers, Distillation Towers, Evaporators, Multiple Effect Evaporators, Performance Assessment of Heat Exchangers, Good Practices in Heat Exchangers, Pinch Analysis

9.1 Heat Transfer Basics

Heat transfer is the transition of thermal energy from a hotter object to a cooler object.

Heat flows from a hot or warm medium to a cold medium in three ways:

- ***By radiation*** from a warm surface to a cooler surface through air space
- ***By conduction*** through solid or fluid materials
- ***By convection*** which involves the physical movement of air

Criteria for heat transfer are

- Difference in temperature between the hot and cold fluids, greater the difference- higher the rate at which the heat will flow between them.
- Area of the conductive barrier provided for heat flow. The greater the area, the larger the amount of heat that will flow in a given time with a given temperature difference.
- Rate of heat flow into, or, out, from each of the fluids. High resistance to heat flow in either fluid will produce a slow overall rate of transfer.
- Flow of heat through the conductive barrier between fluids. Material chosen has to be compatible with the fluids of the process, it must not corrode or contaminate the fluids.

9.2 Concept of Heat Exchangers

A heat exchanger is a device which allows transfer of heat energy from one fluid to another fluid, generally without mixing the two different fluids. Regardless of the function that a heat exchanger fulfills, to transfer heat, the fluids involved must be at different temperatures and they must come into thermal contact, since heat can flow only from the hotter to the cooler fluid.

A heat exchanger can serve any of the following purposes

1. To heat a cooler fluid by means of a hotter fluid
2. To reduce the temperature of a hot fluid by means of a cooler fluid
3. To boil a liquid by means of a hotter fluid
4. To condense a gaseous fluid by means of a cooler fluid
5. To boil a liquid while condensing a hotter gaseous fluid

Heat Exchanger Theory

The heat gained or lost by a fluid in a heat exchanger is given by the following formula:

$Q = m \times Cp \times \Delta T$

Where,

Q = heat transferred (kW)
m = mass flow of the fluid (kg/s)
C_p = specific heat of the fluid (kJ/kg/°C)
ΔT = Temperature difference between inlet and outlet of a fluid (°C)

The basic concept of a heat exchanger is based on the principle that the loss of heat on the high temperature side is exactly the same as the heat gained by the low temperature side. This is expressed by the following formula:

$Q_h = m_h \times Cp_h \times \Delta T_h$

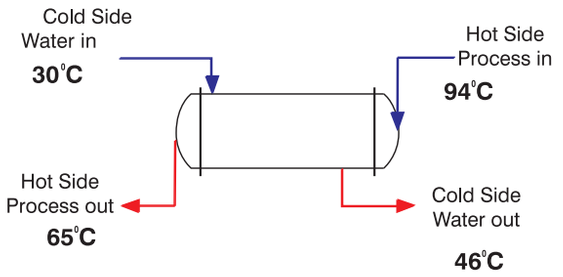
$Q_c = m_c \times Cp_c \times \Delta T_c$

Where subscript "h" = hot fluid, subscript "c" = cold fluid

In theory, the heat given up by the hot fluid is never exactly equal to the heat gained by the cold fluid due to environmental heat losses. However, in practice they are generally assumed to be equal to simplify the calculations involved. Any environmental losses that occur are generally minimized with insulation of equipment and piping.

Example : Calculate the mass flow rate of cooling water required for the heat exchanger shown in figure with the following data:

Mass flow rate of process fluid – 1000 kg/hr
Specific heat of process fluid – 0.9 kcal/kg°C
Specific heat of water – 1 kcal/kg°C



Solution:

$Q_h = m_h \times Cp_h \times \Delta T_h = 1000 \times 0.9 \times (94\text{-}65) = 26{,}100$ kcal/hr

Cooling water flow rate required = 26,100 / [1 x (46-30)]
= 1631 kg/hr

The Heat Exchanger Design Equation

Heat exchanger theory leads to the basic heat exchanger design equation as follows:

$Q = U \times A \times \Delta T_{LMTD}$

Where,
Q is the rate of heat transfer between the two fluids in the heat exchanger in kcal/hr or kJ/hr
U is the overall heat transfer coefficient in kcal/hr m^2 ^oC or $W/m^2 K$
A is the heat transfer surface area in m^2, and
ΔT_{LMTD} is the log mean temperature difference in ^oC, calculated from the inlet and outlet temperatures of both fluids as follows.

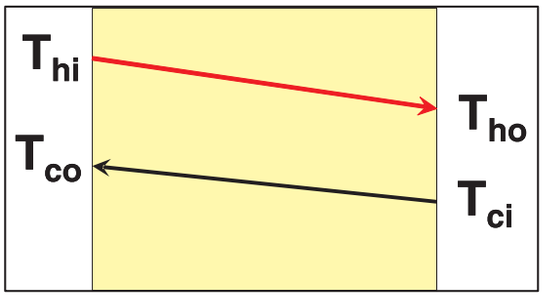


Figure 9.1 Counter Flow Heat Exchanger

$$\Delta T_{LMTD} = Thi \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln \frac{(T_{hi} - T_{co})}{(T_{ho} - T_{ci})}}$$

For design of heat exchangers, the basic heat exchanger design equation can be used to calculate the required heat exchanger area for known or estimated values of the other three parameters, Q, U, and ΔT_{LMTD}. The assumptions that are made in its derivation are that the overall heat transfer coefficient and the specific heat of the fluids remain constant through the heat exchanger.

Log Mean Temperature Difference

The driving force for any heat transfer process is the temperature difference between two fluids. In the heat transfer process, the temperatures of the two fluids keep changing as they pass through the heat exchanger, for which some type of average temperature difference is needed. This average temperature difference through the heat exchanger is described by the log mean temperature difference (LMTD) (Refer Figure 9.2). The larger the temperature difference, the smaller the required heat exchanger area and vice versa.

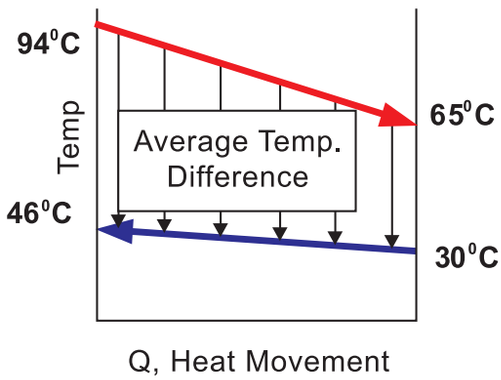


Figure 9.2 LMTD

In Figure 9.1 it appears that the temperature difference between the fluids remains almost constant throughout the heat exchanger. This is rarely the case. In the Figure 9.2 a process stream at 94°C is to be cooled to 65°C using cooling water available at 30°C. It can be observed from the TQ diagram that the temperature difference between the two streams vary widely. This is why the log mean temperature difference is used. In this example the LMTD can be calculated as:

$$\Delta T_{LMTD} = \frac{(94-46)-(65-30)}{\ln\left(\frac{(94-46)}{(65-30)}\right)}$$

$$= 41.2\ ^{o}C$$

There can be special cases where the LMTD equation shown above is not applicable. Consider the case in Figure 9.3. If the LMTD equation is applied to this special case the result would be zero. In this case the LMTD is the same as the temperature difference on each "end" of the heat exchanger (Terminal Temperature Difference), or 30°C.

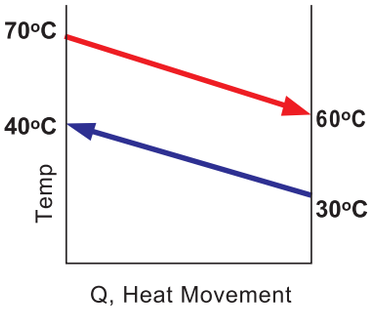


Figure 9.3 Constant ΔT Heat Transfer

Overall Heat Transfer Coefficient, U

The overall heat transfer coefficient describes the rate of heat transfer in the heat exchanger. The overall heat transfer coefficient is a calculated variable based on the physical properties of the fluids involved in the heat transfer (hot and cold) as well as the geometry and type of heat exchanger to be used.

The overall heat transfer coefficient, U, depends on the conductivity through the heat transfer wall separating the two fluids, and the convection coefficients on both sides of the heat transfer wall. Typically in a shell and tube heat exchanger, there would be an inside convective coefficient for the tube side fluid and an outside convective coefficient for the shell side fluid.

Generically, it is described by the following equation:

$$\frac{1}{U} = \frac{1}{h_H} + \frac{\Delta x}{k} + \frac{1}{h_C} + R_f$$

Where,

U = overall heat transfer coefficient ($W/m^2 K$)
h_H = hot side heat transfer coefficient($W/m^2 K$)
h_C = cold side heat transfer coefficient($W/m^2 K$)
Δx = exchanger wall thickness, m
k = exchanger wall material thermal conductivity (W/m K)
R_f = fouling coefficient ($m^2 K/W$)

The heat transfer coefficient for a given heat exchanger is often determined empirically by measuring all of the other parameters in the basic heat exchanger equation and calculating U.

The Temperature Correction Factor, F

The temperature correction factor, F, is used to correct the log mean temperature difference for heat exchangers that lack truly counter-current flow. Many different heat transfer technologies lack truly counter-current flow patterns as a result of their inherent mechanical design. Generally, the value of F lies between 0.85 and 0.98. There are cases when this value can be taken as one, but only if the flow in the exchanger is purely counter-current. The charts are available to look up the temperature correction factor for a given configuration. A method for numerical calculation of F is given in Chapter – 4 of Book – 4.

9.3 Heat Exchanger Types (by flow design)

The flow pattern of the two fluids in a heat exchanger can vary depending on the heat exchanger design and type. Typically the process conditions, available space and economics will define the type and configuration of the heat exchanger. According to the type of flow pattern heat exchangers can be broadly classified as counter flow, parallel flow and cross flow.

Counterflow:

The most effective configuration from a heat transfer standpoint is a counter flow design (also called counter current). In a counter flow arrangement the hot fluid and cold fluid move in opposite directions (Refer Figure 9.4 & 9.5), such that, at one terminal point; the hot side inlet temperature (hottest) is in contact with the cold side outlet temperature (hottest) and at the other terminal point; the cold side inlet temperature (coldest) is in contact with the hot side outlet temperature (coldest). Counter flow designs are most valuable when there is a temperature cross. A temperature cross occurs when the desired outlet temperature of one fluid is between the inlet and outlet temperatures of the other fluid. A true counter flow design is difficult to achieve or impractical for many common heat exchanger applications. However, close to true counter flow could be obtained using plate heat exchangers, double pipe exchangers and spiral heat exchangers.

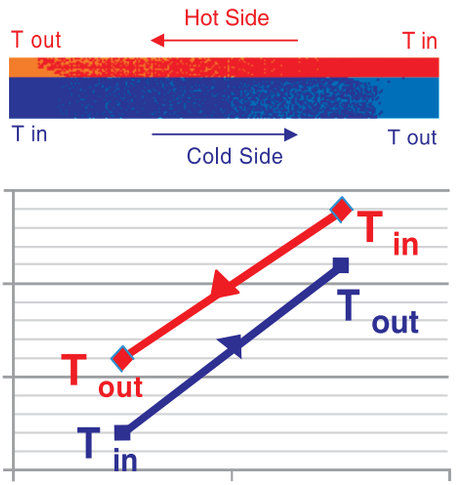
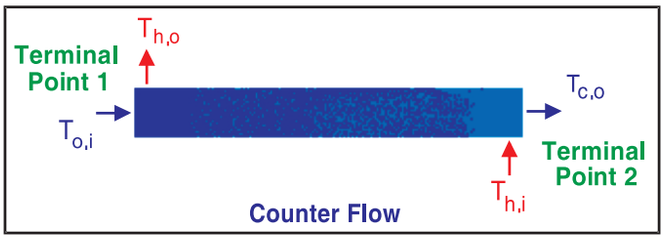


Figure 9.4 Counter Flow Heat Exchanger

$$[LMTD]_{\text{Counter flow}} = \frac{\Delta T_1 - \Delta T_2}{Ln[\Delta T_1 / \Delta T_2]}$$

Where, ΔT1 and ΔT2 are the temperature differences at the two terminal points.



$$\Delta T_1 = T_{h,o} - T_{c,i} \text{ and } \Delta T_2 = T_{h,i} - T_{c,o}$$

Figure 9.5 Counter FLow

Parallel flow

The reverse configuration to counter flow would be parallel flow, where the hot fluid and cold fluid move in the same direction (Figure 9.6 & 9.7), such that, at one terminal point; the hot side inlet temperature (hottest) is in contact with the cold side inlet temperature (coldest), the two fluid progress through the heat exchanger and at the other terminal point; the cold side outlet temperature (hottest) is in contact with the hot side outlet temperature (coldest). A temperature cross cannot be achieved with a parallel flow exchanger. Parallel flow configuration is frequently used in applications where it is desired that the two fluids have close to the same exiting temperatures.

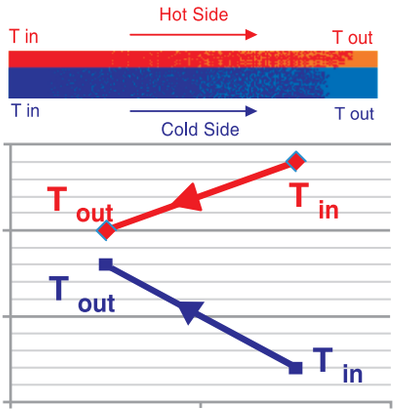
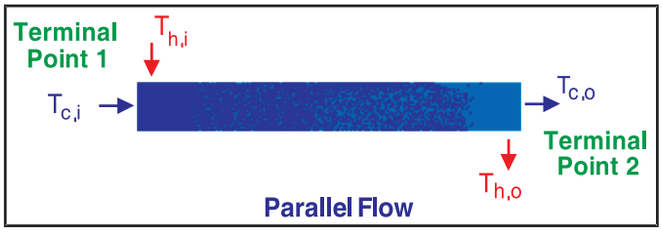


Figure 9.6 Parallel Flow

$$[\text{LMTD}]_{\text{Parallel flow}} = \frac{\Delta T_1 - \Delta T_2}{\text{Ln}[\Delta T_1 / \Delta T_2]}$$

Where, ΔT_1 and ΔT_2 are the temperature differences at the two terminal points.



$$\Delta T_1 = T_{h,i} - T_{c,i} \text{ and } \Delta T_2 = T_{h,o} - T_{c,o}$$

Figure 9.7 Parallel Flow

Cross flow

Most shell and tube heat exchangers and air-cooled heat exchangers are of cross flow design because it can sometimes be impractical to use a counter flow design due to high flow volumes or space restrictions. The flow patterns of two fluids in a cross flow heat exchanger are at right angles to each other (Figure 9.8).

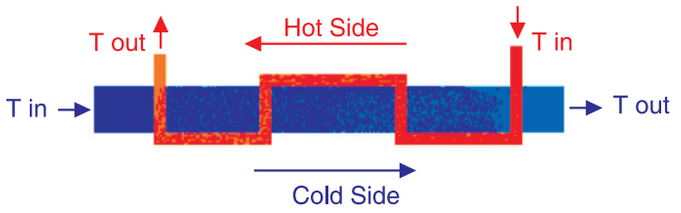


Figure 9.8 Cross Flow

In most applications, at least one fluid makes multiple passes so there is a combination of cross flow and counter flow occurring. With the thermal advantage gained from a multi-pass configuration a temperature cross can be produced in cross flow heat exchanger but not to the same degree as in a true counter flow configuration.

$$[LMTD]_{\text{Cross flow}} = F \times [LMTD]_{\text{Counter flow}}$$

Where, 'F' is the temperature correction factor mentioned earlier.

9.4 Heat Exchanger Types (by construction)

Double Pipe heat Exchangers

The double pipe heat exchanger is probably one of the simplest configurations found in applications. It consists of two concentric circular tubes with one fluid flowing inside the inner tube and the other fluid flowing inside the annular space between the tubes as shown in figure 9.9. Its primary uses are in cooling process fluids where small heat transfer areas are required. It may be designed in a number of arrangements such as parallel flow and counter flow, and combined in series or parallel arrangements with other heat exchangers to form a system.

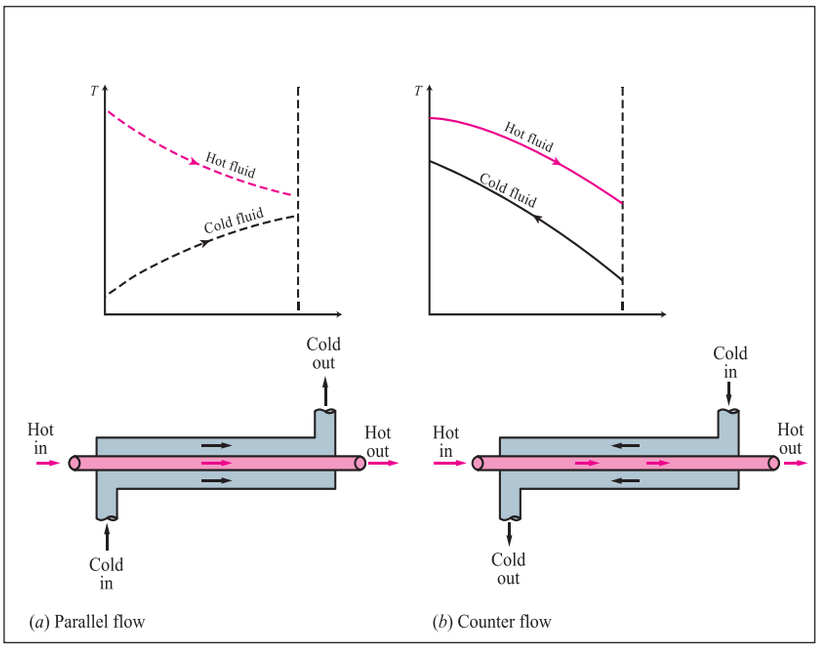


Figure 9.9 Double Pipe Heat Exchanger

Shell and Tube heat Exchangers

Shell-and-tube heat exchangers (Figures 9.10) consist of a bundle of parallel tubes that provide the heat-transfer surface separating the two fluid streams. The tube-side fluid passes axially through the inside of the tubes; the shell-side fluid passes over the outside of the tubes. Baffles external and perpendicular to the tubes direct the flow across the tubes and provide tube support.

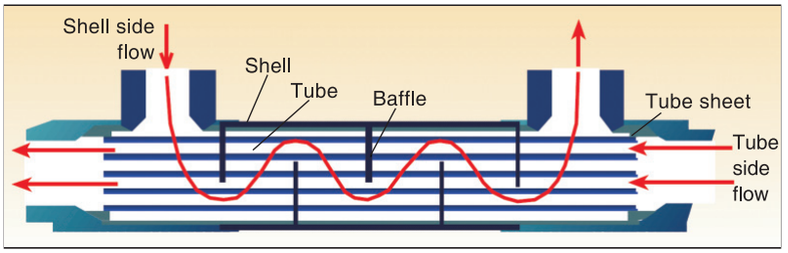


Figure 9.10 Shell and Tube Heat Exchangers

Tube sheets seal the ends of the tubes, ensuring separation of the two streams. The process fluid is usually placed inside the tubes for ease of cleaning or to take advantage of the higher pressure capability inside the tubes. The thermal performance of such an exchanger usually surpasses a coil type but is less than a plate type. Pressure capability of shell-and-tube exchangers is generally higher than a plate type but lower than a coil type.

Plate Heat Exchangers

Plate heat exchangers (Figure 9.11) consist of a stack of parallel thin plates that lie between heavy end plates. Each fluid stream passes alternately between adjoining plates in the stack, exchanging heat through the plates. The plates are corrugated for strength and to enhance heat transfer by directing the flow and increasing turbulence. These exchangers have high heat-transfer coefficients and area, the pressure drop is also typically low, and they often provide very high effectiveness. However, they have relatively low pressure capability.

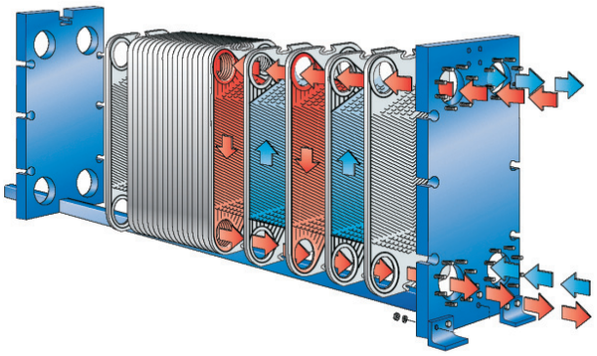


Figure 9.11 Plate Heat Exchangers

Perhaps the biggest advantage of the plate and frame heat exchanger, and a situation where it is most often used, is when the heat transfer application calls for the cold side fluid to exit the exchanger at a temperature significantly higher than the hot side fluid exit temperature. This situation is best explained with another set of T-Q diagrams

Duty 1 shown in the Figure 9.12 (A) can be easily accomplished in a simple shell and tube heat exchanger.

Duty 2 shown in the Figure 9.12 (B) has a severe "temperature cross" or the cold side fluid exiting higher than the hot side fluid. This would require several shell and tube exchangers in series due to the lack of purely counter-current flow. On the other hand, this duty is easily accomplished in a single plate and frame heat exchanger.

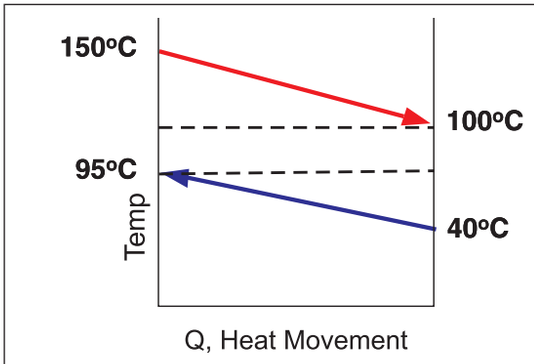


Figure 9.12 (A) Duty 1

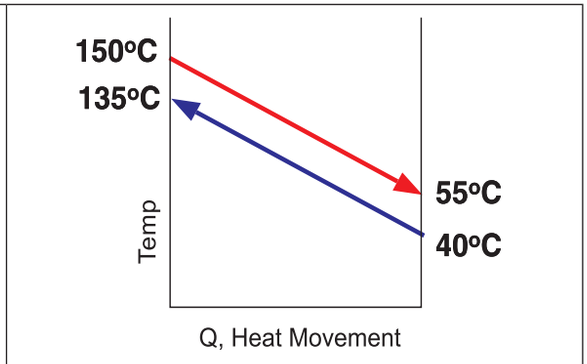


Figure 9.12 (B) Duty 2

The overall heat transfer coefficient of plate heat exchangers under favorable circumstances can be as high as 8,000 W/m^2 ºC. With traditional shell and tube heat exchangers, the U-value will be below 2,500 W/m^2 ºC.

Applications for Counter and Parallel Flows

From the foregoing analysis it can be observed that there are two advantages for counter flow, (a) larger effective LMTD and (b) greater potential energy recovery. The advantage of the larger LMTD, as seen from the heat exchanger equation, is that a larger LMTD permits a smaller heat exchanger area, A, for a given heat transfer, Q. This would normally be expected to result in smaller, less expensive equipment for a given application.

Sometimes, however, parallel flows are desirable (a) where the high initial heating rate may be used to advantage and (b) where it is required the temperatures developed at the tube walls are moderate. In heating very viscous fluids, parallel flow provides for rapid initial heating and consequent decrease in fluid viscosity and reduction in pumping requirement. In applications where moderation of tube wall temperatures is required, parallel flow results in cooler walls. This is especially beneficial in cases where the tubes are sensitive to fouling effects which are aggravated by high temperature.

Effectiveness:

Although heat exchangers are commonly specified only with desired temperatures, the heat-transfer rate is the prime criterion. An exchanger's effectiveness (ε) is the ratio of the actual heat transferred to the heat that could be transferred by an exchanger of infinite size. Effectiveness is the best way to compare different types of heat exchangers. For example, (Figure 9.13) shows a hot-fluid stream being cooled by a cold-fluid stream in a counterflow heat exchanger. When the hot stream exits the exchanger, it must be warmer than the inlet temperature of the cold stream. In an ideal heat exchanger, with

έ = 1, the outgoing hot stream's temperature equals the incoming cold stream's temperature. In addition, this heat exchanger's cold stream exits at a temperature lower than the inlet temperature of the hot stream. The heat-balance equation can be applied to this problem as:

$$\varepsilon = \frac{(\dot{m}\, c_p)_{\text{hot}}(T_{\text{in}} - T_{\text{out}})_{\text{hot}}}{(\dot{m}\, c_p)_{\text{min}}(T_{\text{in hot}} - T_{\text{in cold}})}$$

$$= \frac{(\dot{m}\, c_p)_{\text{cold}}(T_{\text{out}} - T_{\text{in}})_{\text{cold}}}{(\dot{m}\, c_p)_{\text{min}}(T_{\text{in hot}} - T_{\text{in cold}})}$$

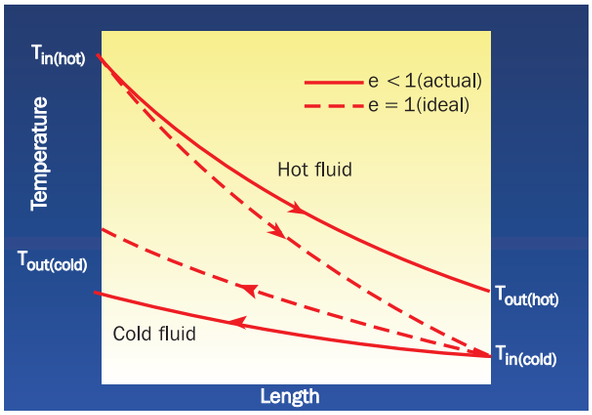


Figure 9.13 Effectiveness

Example - Cooling of milk in a pipe heat exchanger

Milk is flowing into a pipe cooler and passes through a tube of 2.5 cm internal diameter at a rate of 0.4 kg/s. Its initial temperature is 49 °C and it is wished to cool it to 18°C using a stirred bath of constant 10 °C water round the pipe. What length of pipe would be required? Assume an overall coefficient of heat transfer from the bath to the milk of 900 J/m^2 s °C, and that the specific heat of milk is 3890 J/kg °C.

Now

$$q = c_p G\,(T_1 - T_2)$$
$$= 3890 \text{ x } 0.4 \text{ x } (49 - 18)$$
$$= 48{,}240 \text{ J/s}$$

Also $q = UA\Delta T_m$

$$\Delta T_m = [(49 - 10) - (18 - 10)] / \ln [(49 - 10)1(18 - 10)]$$
$$= 19.6°\text{C}.$$

Therefore $48{,}240 = 900 \text{ x } A \text{ x } 19.6$

$$A = 2.73 \text{ m}^2$$

bu $A = \Pi \text{ x } D \text{ x } L$

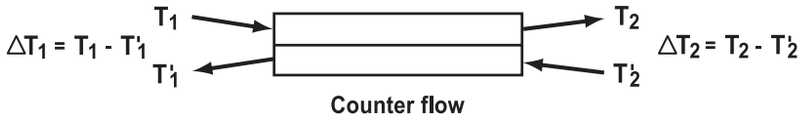
where L is the length of pipe of diameter D

Now $D = 0.025$ m.

$$L = 2.73/(\Delta \text{ x } 0.025)$$
$$= 34.8 \text{ m}$$

Example - Water chilling in a counter flow heat exchanger

In a counter flow heat exchanger, water is being chilled by sodium chloride brine. If the rate of flow of the brine is 1.8 kg/s and that of the water is 1.05 kg/s, estimate the temperature to which the water is cooled if the brine enters at -8 °C and leaves at 10 °C, and if the water enters the exchanger at 32 °C. If the area of the heat-transfer surface of this exchanger is 55 m^2, what is the overall heat-transfer coefficient? Take the specific heats to be 3.38 and 4.18 kJ/kg °C for the brine and the water respectively,



As shown in the figure above, three temperatures are known and the fourth T_{w2} (= T ”$_2$ say) can be found from the heat balance:

By heat balance, heat loss in brine = heat gain in water

1.8 x 3.38 x [10 - (-8)] = 1.05 x 4.18 x (32 - T_{w2})

Therefore T_{w2} = 7°C.

And for counterflow

ΔT_1 = [32 - 10] = 22°C and ΔT_2 = [7 - (-8)] = 15°C.

Therefore ΔT_m = (22 - 15)/ln (22/15)

= 7/0.382

= 18.3°C.

For the heat exchanger

q = heat exchanged between fluids = heat lost by brine = heat gain to water

= heat passed across heat transfer surface

= $UA\Delta T_m$

Therefore

3.38 x 1.8 x 18 = U x 55 x 18.3

U = 0.11 kJ/m^2 s °C

= 110 J/m^2 s °C

9.5 Heat Exchanger Types (by application)

Condensers

In steam applications, condensers (Figure 9.14) are associated with condensing steam turbines and with ejector systems. In steam turbine applications condensers typically operate under a vacuum. They

remove significant amount of latent heat from the exhaust steam allowing it to be recovered as condensate. In steam ejector applications, condensers increase the effectiveness of the ejectors by condensing both the motive steam and condensable pulled from the process, reducing the amount of motive steam required.

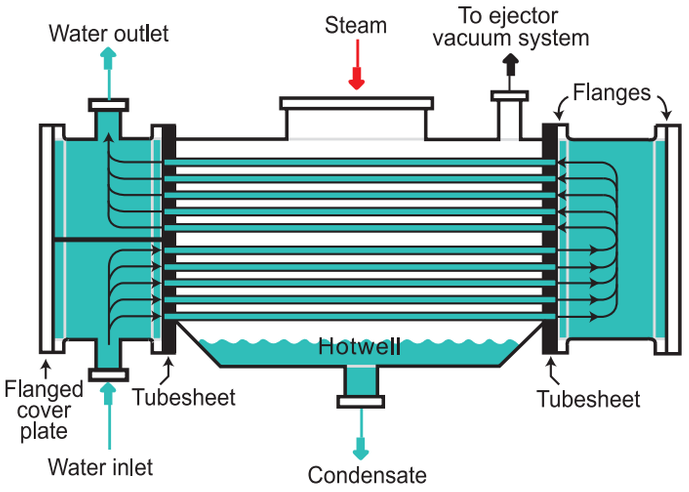


Figure 9.14 Condensers

Condensers can be surface type or barometric. Surface condensers are supplied with cooling water that circulates through condenser tubes providing a cool surface area that causes steam condensation. The condensate is typically collected in a condensate well, and pumped into the condensate return system. Barometric condensers rely on direct contact between the cooling water and the steam. In petroleum refining and chemical manufacturing applications, condensers are also used to condense components from gaseous mixtures. In these applications the condensers use a cooling medium to extract energy from the gases and collect the condensed components.

Distillation Towers

The petroleum refining and chemical manufacturing industries use large amount of steam to facilitate the separation of crude oil or chemical feedstocks into various components. This separation process relies on differences in the boiling points of these hydrocarbon components. Distillation towers (Figure 9.15) use a furnace to heat crude oil above 370°C. As the volatile components boil off and rise up the tower they are cooled and condensed. Steam is injected into bottom of these towers to reduce the partial pressures of the hydrocarbons, which facilitates their separation and to reduce coke formation on tray tower surfaces.

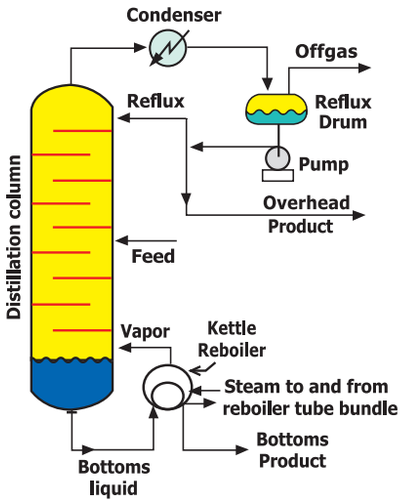


Figure 9.15 Distillation Tower

Evaporators

Evaporators (Figure 9.16) reduce the water content of a liquid generally by heating it with steam, in order to concentrate the product. Evaporators are used extensively in industries such as food processing, chemical, manufacturing, steel, forest products and textiles.

In most cases, evaporators are shell and tube heat exchangers with the steam on the shell side and the product being concentrated in the tubes. Evaporators can be single effect or multiple effects type. A single effect evaporator uses steam at one set of pressure and temperature conditions to boil off the vapor from product. Multiple-effect evaporators take the vapor produced from one evaporator and use it to heat the product in a lower- pressure evaporator.

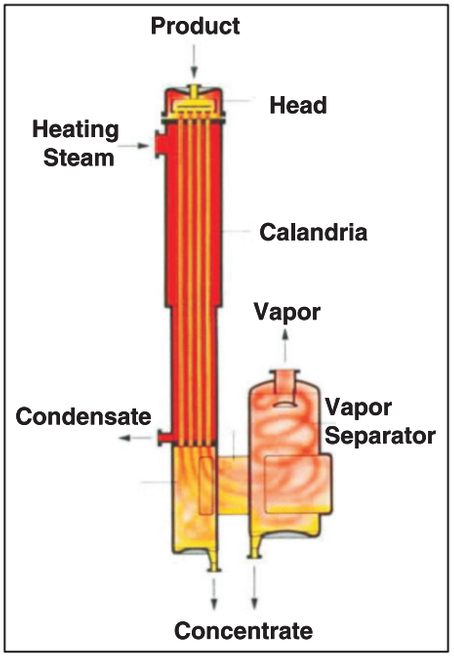


Figure 9.16 Evaporator

Multiple Effect Evaporators

In a multiple effect arrangement, the latent heat of the vapor product off of an effect is used to heat the following effect. Effects are thus numbered beginning with the one heated by steam. It will have the highest pressure. Vapor from Effect I will be used to heat Effect II, which consequently will operate at lower pressure. This continues through the train; pressure drops through the sequence so that the hot vapor will travel from one effect to the next.

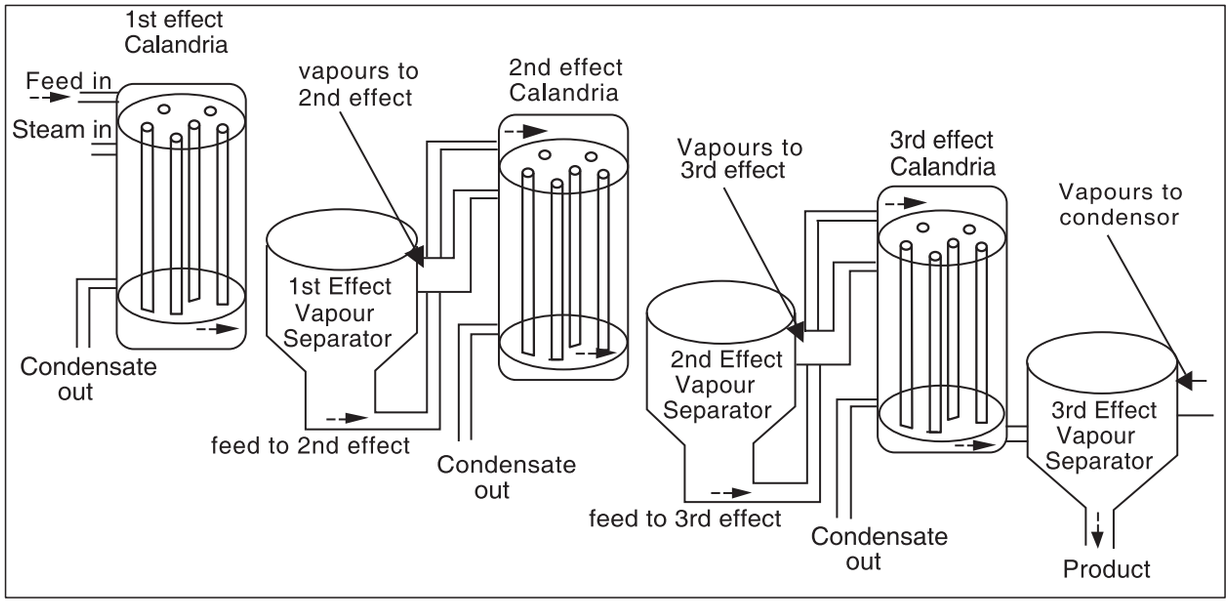


Figure 9.17 Multiple Effect Evaporator

Normally, all effects in an evaporator will be physically the same in terms of size, construction, and heat transfer area. Unless thermal losses are significant, they will all have the same capacity as well.

Unlike single-stage evaporators, these evaporators can be made of up to seven evaporator stages or effects. The energy consumption for single-effect evaporators is very high and makes up most of

the cost for an evaporation system. Putting together evaporators saves heat and thus requires less energy. Adding one evaporator to the original decreases the energy consumption to nearly 50% of the original amount. Adding another effect reduces it to nearly 33% and so on. A heat saving equation can be used to estimate how much one will save by adding a certain amount of effects. The number of effects in a multiple-effect evaporator is usually restricted to seven because after that, the equipment cost starts catching up to the money saved from the energy requirement drop.

Effect of Heat Exchanger efficiency

Efficiency and productivity of many production processes inherently depend on good transfer of heat from the heat transfer surfaces and equipment. The design technical parameters cannot often be achieved unless the heating / cooling processes are carried out efficiently.

A reduction in heat transfer that occurs almost invariably has an impact on cycle time and product cost. To minimize adverse impact, heat exchanger performance should be monitored and maintained at regular intervals that are determined from cost benefit criteria.

A good operation & maintenance policy therefore must be formulated right from its installation and during subsequent years for achieving the desired process efficiency.

9.6 Pinch Analysis & Pinch Technology Application For Process & Energy Efficiency Improvements

What is Pinch ?

Consider the following simple process [Figure 9.18(a)] where feed stream to a reactor is heated before inlet to a reactor and the product stream is to be cooled. The heating and cooling are done by use of steam (Heat Exchanger -1) and cooling water (Heat Exchanger-2), respectively. The Temperature (T) vs. Enthalpy (H) plot for the feed and product streams depicts the hot (Steam) and cold (CW) utility loads when there is no vertical overlap of the hot and cold stream profiles.

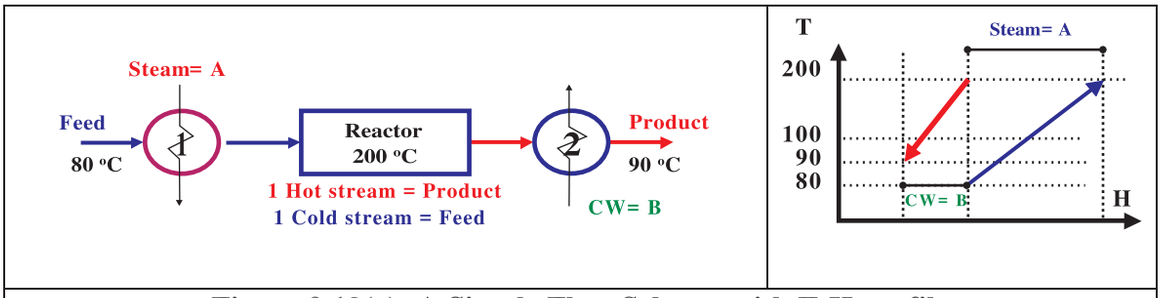


Figure 9.18(a): A Simple Flow Scheme with T-H profile

An alternative, improved scheme is shown in Figure 9.18(b) where the addition of a new 'Heat Exchanger–3' recovers product heat (X) to preheat the feed. The steam and cooling water requirements

also get reduced by the same amount (X). The amount of heat recovered (X) depends on the 'minimum approach temperature' allowed for the new exchanger. The minimum temperature approach between the two curves on the vertical axis is ΔTmin and the point where this occurs is defined as the "pinch". From the T-H plot, the X amount corresponds to a ΔTmin value of 20 °C. Increasing the ΔTmin value leads to higher utility requirements and lower area requirements.

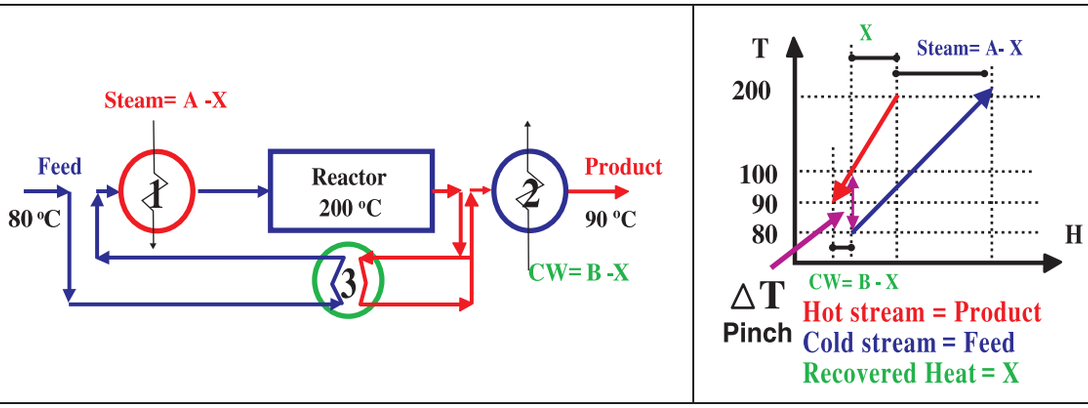


Figure 9.18(b): Improved Flow Scheme with T-H profile

Curves are "shifted "horizontally towards each other till they are vertically apart by 20°C which is the ΔT minimum and this is called the 'Pinch'

Pinch Principle

Energy pinch identifies the minimum cost of hot utility targets, as well as the projects that allow us to reach these targets in practice (or to get close to them).

When considering any pinch-type problem the following principles apply:

- Processes can be defined in terms of supplies and demands (sources and sinks) of commodities (energy, water, etc.).
- The optimal solution is achieved by appropriately matching suitable sources and sinks.
- The defining parameter in determining the suitability of matches is quality, ie. temperature
- Inefficient transfer of resources means that the optimal solution cannot be achieved. In fact, the amount of inefficient transfer is equal to the wasteful use of imported commodities such as cooling water and steam.

Building the composite curves

One of the principle tools of pinch analysis involving multiple heat exchangers is the graphic representation of composite curves (Figure 9.19), the construction of which is simple but powerful. Composite curves are used to determine the minimum energy-consumption target for a given process. The curves are profiles of a process' heat availability (hot composite curve) and heat demands (cold

composite curve). The degree to which the curves overlap is a measure of the potential for heat recovery, as illustrated in Figure 9.19.

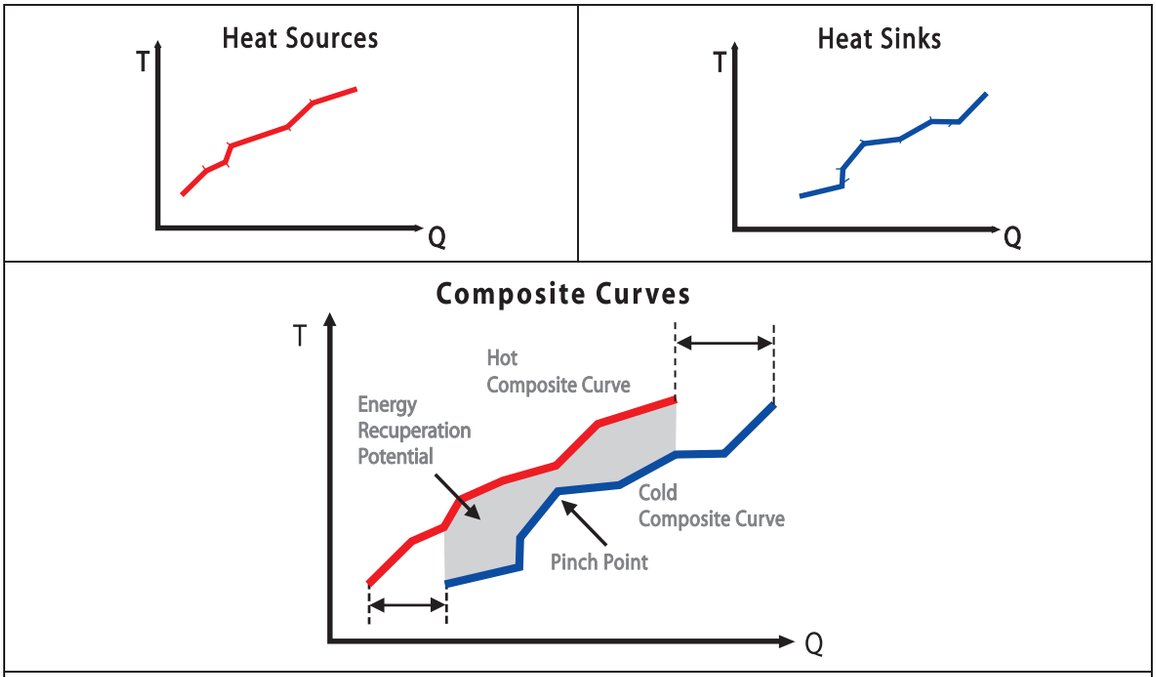


Figure 9.19 Building Composite Curves

Constructing the curves requires only a complete and consistent heat and mass balance of the process in question. Data from the heat and mass balance are firstused to define process streams in terms of their temperature and heating or cooling requirements

Building Composite Curves

The following are the data of heat exchangers from a process

Stream	Stream type	Supply Temperature (°C)	Target Temperature (°C)	Duty (kW)	CP (kW/°C)
1	Hot	200	100	2000	20
2	Hot	150	60	3600	40
3	Cold	80	120	3200	80
4	Cold	50	220	2550	15

There are two hot streams (Streams 1 and 2) and two cold streams (Streams 3 and 4). Note that CP is defined as (mass flow rate) x (specific heat) and is represented by the slope of the curve in Figure 9.20 (a). For example, Stream 1 is cooled from 200°C to 100°C, releasing 2000 kW of heat, and so has a CP of 20 kW/°C.

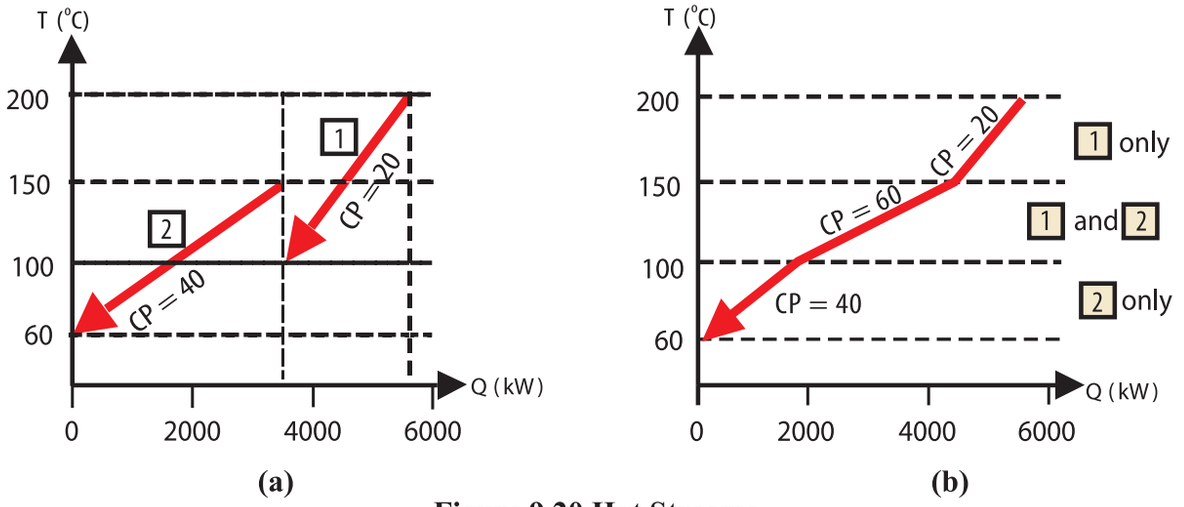


Figure 9.20 Hot Streams

Figure 9.20 (a) shows the hot streams plotted separately on a temperature-duty diagram. The hot composite curve in Figure 9.20 (b) is constructed by simply adding the enthalpy changes of the individual streams within each temperature interval. In the temperature interval 200°C to 150°C, only Stream 1 is present. Therefore, the CP of the composite curve equals the CP of Stream 1, i.e., 20. In the temperature interval 150°C to 100°C, both Streams 1 and 2 are present. Here, the CP of the hot composite is the sum of the CPs of the two streams, i.e., 20+40=60. In the temperature interval 100°C to 60°C, only Stream 2 is present, so the CP of the composite is 40.

Construction of the cold composite curve is done similarly to that of the hot composite curve, but combines the temperature enthalpy curves of the cold streams (see Figures 9.21 (c) and (d)).

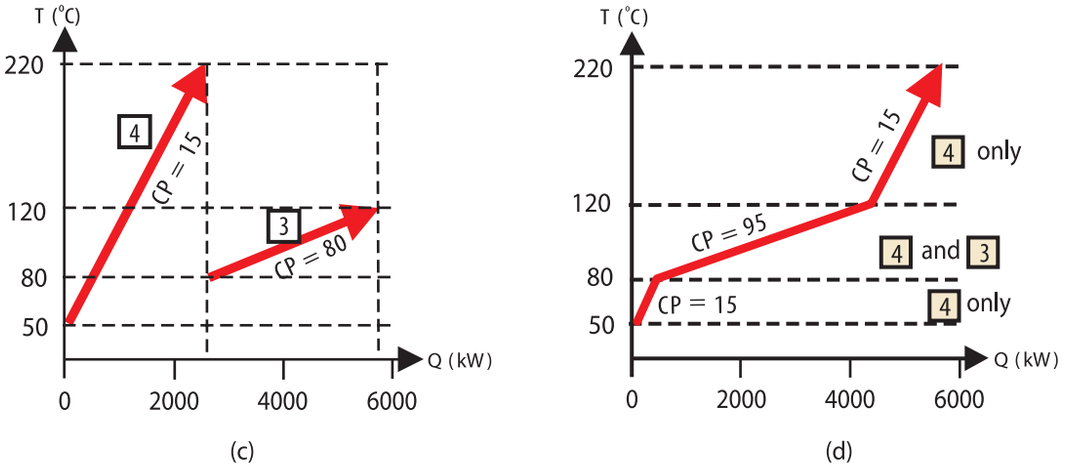


Figure 9.21 Hot Streams

Determining the energy targets

To determine the minimum energy target for the process, the cold composite curve is progressively moved toward the hot composite curve, as shown in Figure 9.22 (a). Note that the enthalpy axis

measures relative quantities, which means that we are representing the enthalpy change of process streams. Moving a composite curve horizontally does not, in any way, change the stream data. The closest approach of the curves is defined by the minimum allowable temperature difference, ΔT_{min} (Figure 9.22 (a)). This value determines the minimum temperature difference that will be accepted in a heat exchanger. A value of 10°C has been used in this example.

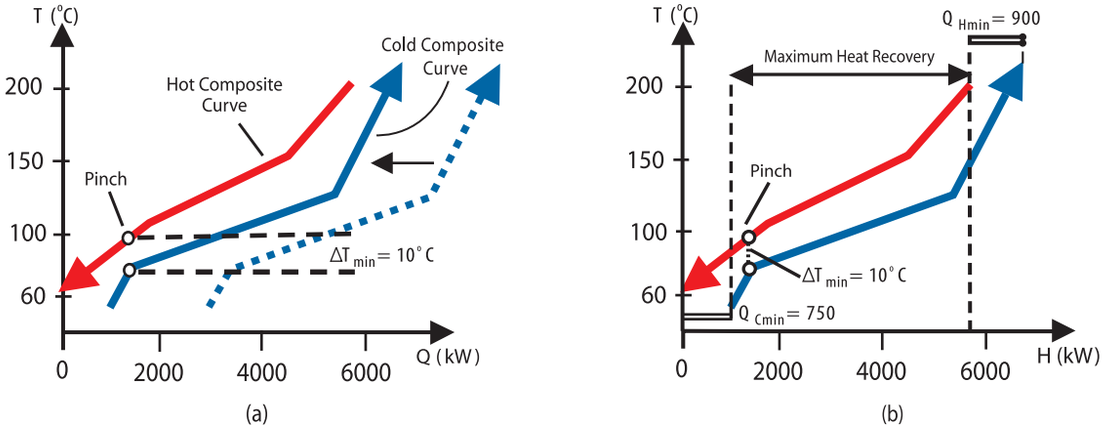


Figure 9.22 Determining the Energy Targets

The overlap between the composite curves shows the maximum possible process-heat recovery (as shown in Figure 9.22 (b)), indicating that the remaining heating and cooling needs are the minimum hot utility requirement (QHmin) and the minimum cold utility requirement (QCmin) of the process for the chosen ΔTmin. In this example, the minimum hot utility (QHmin) is 900 kW and the minimum cold utility (QCmin) is 750 kW.

Using pinch analysis, targets can be set for minimum energy consumption prior to any heat exchanger network design. This enables quick identification of the scope for energy savings, at an early stage of the analysis. This single benefit is likely to be the biggest strength of the approach.

Benefits and Applications of Pinch Technology

One of the main advantages of Pinch Technology over conventional design methods is the ability to set energy and capital cost targets for an individual process or for an entire production site ahead of design. Therefore, in advance of identifying any projects, the scope for energy savings and investment requirements can be determined.

General Process Improvements

In addition to energy conservation studies, Pinch Technology enables process engineers to achieve the following general process improvements:

Update or Modify Process Flow Diagrams (PFDs): Pinch quantifies the savings available by changing the process itself. It shows where process changes reduce the overall energy target, not just local energy consumption.

Conduct Process Simulation Studies: Pinch replaces the old energy studies with information that can be easily updated using simulation. Such simulation studies can help avoid unnecessary capital costs by identifying energy savings with a smaller investment before the projects are implemented.

Set Practical Targets: By taking into account practical constraints (difficult fluids, layout, safety, etc.), theoretical targets are modified so that they can be realistically achieved. Comparing practical with theoretical targets quantifies opportunities "lost" by constraints - a vital insight for long-term development.

Debottlenecking: Pinch Analysis, when specifically applied to debottlenecking studies, can lead to the following benefits compared to a conventional revamp:

- Reduction in capital costs
- Decrease in specific energy demand giving a more competitive production facility

For example, debottlenecking of distillation columns by Column Targeting can be used to identify alternate, cost effective options to column re-traying or installation of a new column.

Determine Opportunities for Combined Heat and Power (CHP) Generation: A well-designed CHP system significantly reduces power costs. Pinch shows the best type of CHP system that matches the inherent thermodynamic opportunities on the site. Unnecessary investments and operating costs can be avoided by sizing plants to supply energy that takes heat recovery into consideration. Heat recovery should be optimized by Pinch Analysis before specifying CHP systems.

Decide what to do with low-grade waste heat: Pinch shows, which waste heat streams, can be recovered and lends insight into the most effective means of recovery.

Solved Example:

A counter-flow double pipe heat exchanger using hot process liquid is used to heat water, which flows at 10.5m^3/hr. The process liquid enters the heat exchanger at 180°C and leaves at 130°C. The inlet and exit temperature of water are 30°C and 90°C respectively. Specific heat of water is 4.18 kJ/kg°C.

a) Calculate the heat transfer area, if overall heat transfer coefficient is 814 W/m^2°C.
b) What would be the percentage increase in are, if the fluid flows were parallel?

Ans:

Water flow rate = 10.5 x 1000 = 10500 kg/hr

Heat content in water = m x Cp x ΔT
= 10500 x 4.18 x (90 – 30)
= 2633400 kJ/hr
= 2633400 / 3600
= 731.5 kW

For Counter current flow:

$\Delta T1 = 180 - 90 = 90°C$
$\Delta T2 = 130 - 30 = 100°C$

LMTD of counter flow = (100-90)/ ln (100/90) **= 95°C**

Overall heat transfer coeff. = 814 W/m^2°C

Area of heat exchanger for counter flow = 731.5 x 1000 / (814x 95)
= 9.5 m^2

For Parallel flow:

$\Delta T1 = 180 - 30 = 150°C$
$\Delta T2 = 130 - 90 = 40°C$

LMTD of parallel flow = (150-40)/ ln(150/40) = 83°C

Overall heat transfer coeff. = 814 W/m^2°C

Area of heat exchanger for parallel flow = 731.5 x 1000 / (814 x 83)
= 10.8 m^2

Increase in the area for parallel flow = [(10.8 – 9.5) / 9.5] x 100
= 14 %

QUESTIONS	
Objective Type Questions	
1.	The criteria for heat transfer among the following is the difference in _____ between the hot and cold fluid. a) temperature b) pressure c) density d) viscosity
2.	What is the unit of overall heat transfer coefficient? a) W/m^2 K b) W/m^3 K c) W^2/m^2 K d) W^2/m^3 K
3.	Pinch analysis uses the _____ law of thermodynamics. a) first b) second c) both d) none
4.	Correction factor for LMTD is applicable for a) counter current b) parallel flow c) cross flow d) both a & b
5.	If the terminal temperature differences at the hot and cold end of a heat exchanger is same, then the LMTD is a) 100 ^{0}C b) 0 ^{0}C c) 50 ^{0}C d) none of the above
6.	Higher the heat transfer coefficient ---------- a) lower will be the area b) higher will be the area c) lower will be the heat transfer d) higher will be the LMTD
7.	1000 kg of water is to be heated from 30 to 70 ^{0}C. the heat gained by the water will be a) 70,000 kcal b) 40,000 kcal c) 40,000 kJ d) 100,000 kcal
8.	The effectiveness of a heat exchanger does not depend on a) specific heat of hot fluid b) specific heat of cold fluid c) inlet temperature of hot fluid d) LMTD
9.	A pinch analysis can result in a) reduction in cooling water b) reduction in steam c) increase in cooling water d) both a & b
10.	Select the odd one among the following a) condenser b) distillation column c) evaporator d) cooling tower
Short Type Questions	
S-1	Explain the significance of LMTD
S-2	In a heat exchanger 90^{o}C hot water flows inside the tube. Hot combustion gas at 600^{o}C passes at the outside of the water tube. What is your best judgement about the surface temperature of the tube at the gas side? Explain.
S-3	In a high temperature furnace recuperator, explain why parallel flow is adopted.
S-4	Explain the significance of higher and lower pinch
S-5	Explain what is meant by effectiveness

Long Type Questions	
L-1	Explain with a sketch how a multiple effect evaporator reduces steam consumption when compared to single effect evaporator.
L-2	Estimate the steam requirement as you start to heat 50 kg of pea soup in a jacketed pan, if the initial temperature of the soup is 18°C and the steam used is at 100 kPa gauge. The pan has a heating surface of 1 m^2 and the overall heat transfer coefficient is assumed to be 300 J/m^2 s °C. From steam tables saturation temperature of steam at 100 kPa gauge = 120°C and latent heat = 2202 kJ/kg.

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ANNEXURE

CHECKLISTS & TIPS FOR ENERGY EFFICIENCY IN THERMAL UTILITIES

Boilers

- Preheat combustion air with waste heat.
  (22 ^{0}C reduction in flue gas temperature increases boiler efficiency by 1%)
- Use variable speed drives on large boiler combustion air fans with variable flows.
- Burn wastes if permitted.
- Insulate exposed heated oil tanks.
- Clean burners, nozzles, strainers, etc.
- Inspect oil heaters for proper oil temperature.
- Close burner air and/or stack dampers when the burner is off to minimize heat loss up the stack.
- Improve oxygen trim control (e.g. -- limit excess air to less than 10% on clean fuels).
  (5% reduction in excess air increases boiler efficiency by 1% or: 1% reduction of residual oxygen in stack gas increases boiler efficiency by 1%)
- Automate/optimize boiler blowdown. Recover boiler blowdown heat.
- Use boiler blowdown to help warm the back-up boiler.
- Optimize deaerator venting.
- Inspect door gaskets.
- Inspect for scale and sediment on the water side.
  (A 1 mm thick scale (deposit) on the water side could increase fuel consumption by 5 to 8%.)
- Inspect for soot, flyash, and slag on the fire side.
  (A 3 mm thick soot deposition on the heat transfer surface can cause an increase in fuel consumption to the tune of 2.5%)
- Optimize boiler water treatment.
- Add an economizer to preheat boiler feedwater using exhaust heat.
- Recycle steam condensate.
- Study part-load characteristics and cycling costs to determine the most-efficient mode for operating multiple boilers.
- Consider multiple or modular boiler units instead of one or two large boilers.
- Establish a boiler efficiency-maintenance program. Start with an energy audit and follow-up, then make a boiler efficiency-maintenance program a part of your continuous energy management program.

Steam System

- Fix steam leaks and condensate leaks.
  (A 3 mm diameter hole on a pipe line carrying 7 Kg/cm^2 steam would waste 33 Kilo litres of fuel oil per year)
- Accumulate work orders for repair of steam leaks that can't be fixed during the heating season due to system shutdown requirements. Tag each such leak with a durable tag with a good description.
- Use back pressure steam turbines to produce lower steam pressures.
- Use more-efficient steam desuperheating methods.
- Ensure process temperatures are correctly controlled.
- Maintain lowest acceptable process steam pressures.
- Reduce hot water wastage to drain.
- Remove or blank off all redundant steam piping.
- Ensure condensate is returned or re-used in the process.
  (6^{0}C raise in feed water temperature by economiser/condensate recovery corresponds to a 1% saving in fuel consumption, in boiler)
- Preheat boiler feed-water.
- Recover boiler blowdown.
- Check operation of steam traps.
- Remove air from indirect steam using equipment
  (0.25 mm thick air film offers the same resistance to heat transfer as a 330 mm thick copper wall)
- Inspect steam traps regularly and repair malfunctioning traps promptly.
- Consider recovery of vent steam (e.g. -- on large flash tanks).
- Use waste steam for water heating.
- Use an absorption chiller to condense exhaust steam before returning the condensate to the boiler.
- Use electric pumps instead of steam ejectors when cost benefits permit
- Establish a steam efficiency-maintenance program. Start with an energy audit and follow-up, then make a steam efficiency-maintenance program a part of your continuous energy management program.

Furnaces

- Check against infiltration of air: Use doors or air curtains
- Monitor O_2 /CO_2/CO and control excess air to the optimum level
- Improve burner design, combustion control and instrumentation.
- Ensure that the furnace combustion chamber is under slight positive pressure

- Use ceramic fibres in the case of batch operations
- Match the load to the furnace capacity
- Retrofit with heat recovery device
- Investigate cycle times and reduce
- Provide temperature controllers
- Ensure that flame does not touch the stock

Insulation

- Repair damaged insulation.
  (A bare steam pipe of 150 mm diameter and 100 m length, carrying saturated steam at 8 kg/cm^2 would waste 25,000 litres furnace oil in a year)
- Insulate any hot or cold metal or insulation.
- Replace wet insulation.
- Use an infrared gun to check for cold wall areas during cold weather or hot wall areas during hot weather.
- Ensure that all insulated surfaces are cladded with aluminum
- Insulate all flanges, valves and couplings
- Insulate open tanks
  (70% heat losses can be reduced by floating a layer of 45 mm diameter polypropylene (plastic) balls on the surface of 90^{0}C hot liquid/condensate)

Waste heat recovery

- Recover heat from flue gas, engine cooling water, engine exhaust, low pressure waste steam, drying oven exhaust, boiler blowdown, etc.
- Recover heat from incinerator off-gas.
- Use waste heat for fuel oil heating, boiler feedwater heating, outside air heating, etc.
- Use chiller waste heat to preheat hot water.
- Use heat pumps.
- Use absorption refrigeration.
- Use thermal wheels, run-around systems, heat pipe systems, and air-to-air exchangers.